Petroleum Production Engineering

A
Special Presentation
Conducted by IPCS
for
The Conventional Energy Training Project

July 25 - 29, 1983

Presented
by
Dr. R. Eugene Collins

DAN-9997-C-00-1079-00

IPCS
International Petroleum Consulting Services
1200 New Hampshire Avenue, N.W.
Suite 320
Washington, D.C. 20036 U.S.A.
Telephone Numbers: (202) 331-8214
(202) 331-8215
Telex Number: 248826 WOS

CONTENTS

WELL PERFORMANCE

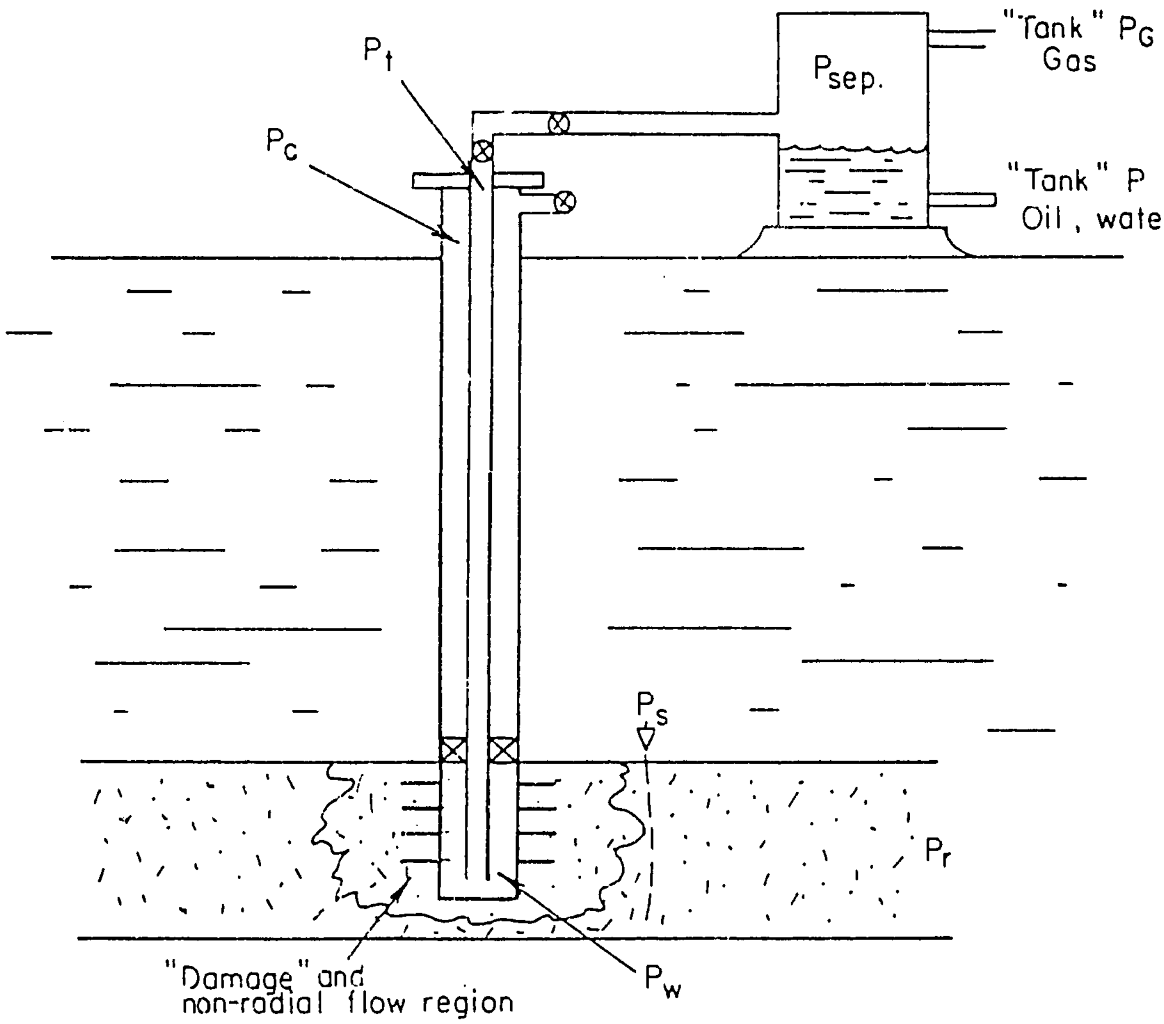
GAS LIFT

SUCKER ROD PUMPS

Well Performance

Factors Affecting Well Productivity

The basic elements determining well productivity, that is flow rate, can be visualized in the sketch below.



The driving "force", or energy for flow is pressure and this is dissipated in two ways, one is work against gravity and the other is work against viscous drag, or "friction". The total drop in pressure from the reservoir, at the "drainage radius" of the well, to the storage tank can be separated into parts thus:

Number	Domain	P
I	Drainage Radius to "skin" of Well	$P_r - P_s$
II	"Skin" of Well to Bottom Hole	$P_s - P_w$
III	Bottom Hole to Tubing Head	$P_w - P_t$
IV	Tubing Head to Separator	$P_t - P_{Sep.}$
V	Separator to Tank	$P_{Sep} - P_o$ or $P_{Sep} - P_G$

These are described as follows:

I $P_r - P_s$ is determined by reservoir properties and drive mechanisms: permeability, porosity, thickness, relative permeabilities, capillary pressures; fluid properties, absolute pressure level and temperature, fluid viscosities, and densities; solution gas, fluid expansion, gas cap, or water drive and flow rate.

II $P_s - P_w$ is determined by formation penetration, flow rate, well bore radius, perforations, formation damage by drilling fluid or completion fluid, or fines migration, stimulation by acid or fracturing, single or multi-phase flow, fluid properties and absolute pressure level and temperature, formation permeability, etc.

III $P_w - P_t$ is determined by depth, fluid viscosities and densities, tubing diameter and roughness, flow rate, pressure level and temperature.

IV $P_t - P_{Sep}$ is determined by fluid properties, flow rate, absolute pressure level and temperature, pipe and choke size, and pipe line length.

V $P_{Sep} - P_o$, or $P_{Sep} - P_G$, is determined by fluid properties , absolute pressure level and temperature, flow rate, pipe diameter, roughness and length.

Well Production Testing

Objectives of Well Testing:

(1) Determine amount and type of fluids produced.

(2) Determine maximum production capability of well.

(3) Determine properties of reservoir, K, ϕ.

(4) Determine reservoir pressure.

(5) Determine need for remedial treatment of well; evaluate well damage.

(6) Determine effectiveness of a well treatment, post facto.

(7) Determine appropriate well equipment to achieve allowable production rate; tubing size, separator, and artificial lift equipment.

Types of Well Tests

1. Periodic production tests; gauging
2. Productivity or Deliverability Tests
3. Inflow performance tests
4. Transient pressure tests.

Periodic-Production Testing; Gauging

This consists of simply measuring the amount and type of fluids produced and is routinely carried out using a gas-oil separator and a stock tank, with a device such as an orifice meter to measure gas flow rate and a hand tape to measure amounts of oil and water in the stock tank. Modern techniques use more sophisticated systems with automatic recording.

Productivity Tests: Oil Wells

The Productivity Index, P I, or J, is defined by

$$P\,I = J = \frac{q}{P_r - P_w}$$

where conventionally

q = Liquid (oil or oil+water) flow rate $V_{(STB/D)}$

P_r = Shut-in well (Reservoir) pressure (Psi)

P_w = Flowing bottom-hole pressure (Psi)

Here for $q = q_o$, oil Pi is defined and for $q = q_o + q_w$, total P I is defined.

P I, or J, is determined by reservoir and well properties. Measurement of P I requires a bottom hole pressure measurement, either directly with a down-hole instrument or indirectly with estimates from surface pressure data, while flowing and when shut-in. The long-term shut-in bottom-hole pressure is a measure of P_r.

Measures of Well Productivity

Productivity Index

The basic measure of well production efficiency is the Productivity Index, or P I, defined above as

$$J = P\,I = \frac{q}{P_r - P_w}$$

where q is production rate measured at surface conditions and $P_r - P_w$ is the "drawdown" at bottom hole. i.e., P_r is essentially shut-in, or static, reservoir pressure and P_w is flowing bottom hole pressure.

Factors affecting the value of P I are shown in part by using Darcy's law and approximating fluid flow as steady-state, incompressible, single-phase flow with pressure P_r at some "drainage radius" r_e on a cylindrical boundary about well axis. For steady-state radial incompressible flow Darcy's law gives

$$q = -2\pi r \frac{kh}{\mu} \frac{\partial P}{\partial r} = \text{constant}$$

This can be used for the domain r_w to r_e, the drainage radius with $P = P_r$ at r_e, to show that

$$J = \frac{q}{P_r - P_w} \approx \frac{0.00708kh}{B\mu \ln \frac{r_e}{r_w}} \frac{1}{1+S_D}$$

Here k(md) is reservoir permeability, h(ft) is producing zone thickness, μ (cp) fluid viscosity and B the fluid formation volume factor. The term S_D is a "catch-all" dimensionless, well damage factor which accounts for non-radial flow near the well-bore and/or damaged or improved permeability near the well. For a completely penetrating well with open-hole completion and a damaged-zone permeability k_s in a skin-zone out to radius r_s

$$S_D = \left(\frac{k}{k_s} - 1\right) \frac{\ln \frac{r_s}{r_w}}{\ln \frac{r_e}{r_w}}$$

This quantity is positive $S_D > 0$ for $k_s < k$ and negative for $k_s > k$. $S_D > 0$ could result from fresh water filtrate from drilling mud entering the formation while $S_D < 0$ could result from acid stimulation treatment.

Clearly two-phase flow and/or compressibility effects prevent any rigorous application of these simple relationships.

The Specific Productivity Index is often used and is the ratio of P I to the thickness, h, of the producing zone, expressed in feet. i.e., STB/D/ft./Psi.

While the Productivity Index is a very useful concept it suffers from being non-constant, that is, the PI measured at one drawdown, $P_r - P_w$, will not have the same value as one measured at another drawdown, even at the same reservoir pressure.

Inflow Performance Test: Oil Wells

It is obvious that the productivity index in a real well will not be a simple constant characteristic of the well and reservoir configuration because of multi-phase flow and compressibility effects. J. V. Vogel (1968) developed on empirical method to account for these effects by solving the material balance equations for radial flow based upon Darcy's law, with some approximations, for "gas" and "oil" flow with gas coming out of solution in the oil. Flow is below the bubble point. Specifically he assumed:

(1) reservoir is circular with closed boundary

(2) well completely penetrates formation

(3) porosity, permeability and thickness uniform and constant

(4) gravity segregation is negligible

(5) capillary pressure negligible

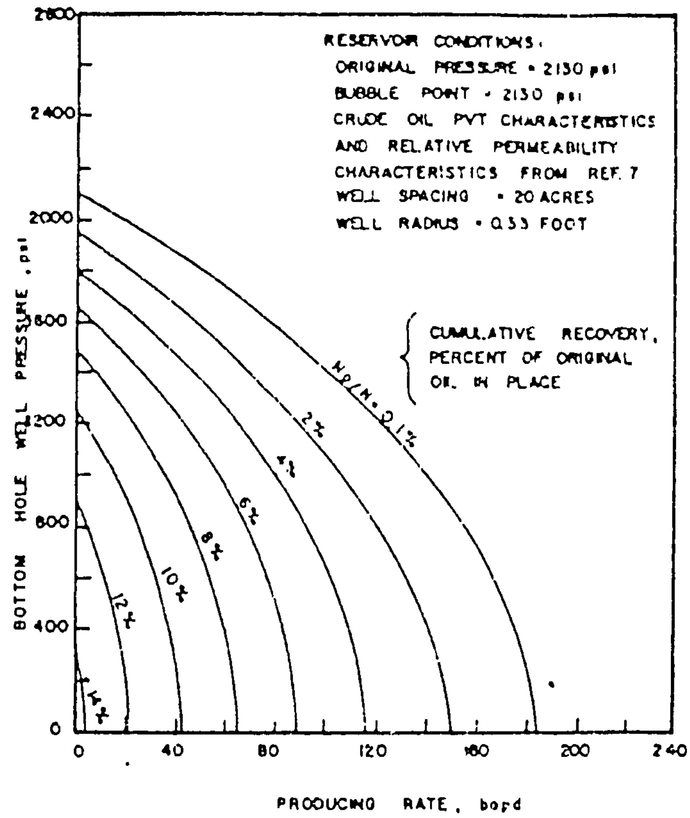
(6) gas-oil in local equilibrium.

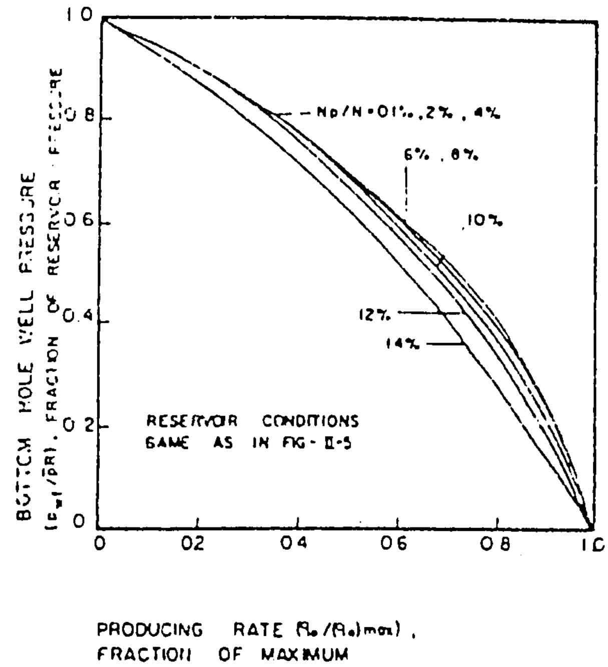
From many runs of the numerical integration he showed that to a very good approximation the graph of stabilized oil flow rate at the well, versus flowing bottom-hole pressure, could be represented by the "universal" dimensionless relationship

$$\frac{q}{q_{max}} = 1 - 0.20\left(\frac{P_w}{P_r}\right) - 0.80\left(\frac{P_w}{P_r}\right)^2$$

Where q_{Max} is the maximum flow rate into the well resulting when bottom-hole pressure is reduced to zero. This is known as the "IPR", or Inflow-Performance -Relationship.

The IPR shows that data of q vs P_w at different values of P_r appears as shown in the following sketch.





Dimensionless inflow performance relationships for a solution gas drive reservoir (after Vogel)

Since P_r naturally declines with cumulative production in a solution-gas-drive reservoir the successive curves here represent well performance at various stages of cumulative production from the reservoir.

As given above this relationship does not include well damage, or "skin", effects. The effect of a "skin" of higher flow resistance near the well is to require a lower bottom-hole pressure, or greater "draw-down", to sustain a given flow rate. Thus the P_w in the above equation is the undamaged P_w and the actual P_w, say P_w' is less by P_s, the pressure drop across the "skin".

The usefulness of the IPR lies in the fact that a measurement of P_r and the flow rate q at one bottom-hole pressure determines the entire curve;

i.e., determines q at any possible P_w while this P_r exists. This of course in the absense of damage. i.e., given P_r, P_w and q, insert in Vogel equation above and solve for q_{Max}; then all coefficients are determined.

The limitation of produc ivity index, J, as a means of predicting the flow capability of a well at some bottom hole pressure other than that used to determine J is shown in the sketch below. The dotted straight line below is represented by

$$P_w = P_r - \frac{1}{J} q$$

and clearly this predicts a P_w not on the Vogel curve except at one point, the point used to determine J. i.e., the intersection with the Vogel curve.

Standing's Forecast of Inflow Performance

Vogel's Equation, using now q_o for oil production rate,

$$q_o = q_m \left(1 - .2 \left(\frac{P_w}{P_r}\right) - .8 \left(\frac{P_w}{P_r}\right)^2\right)$$

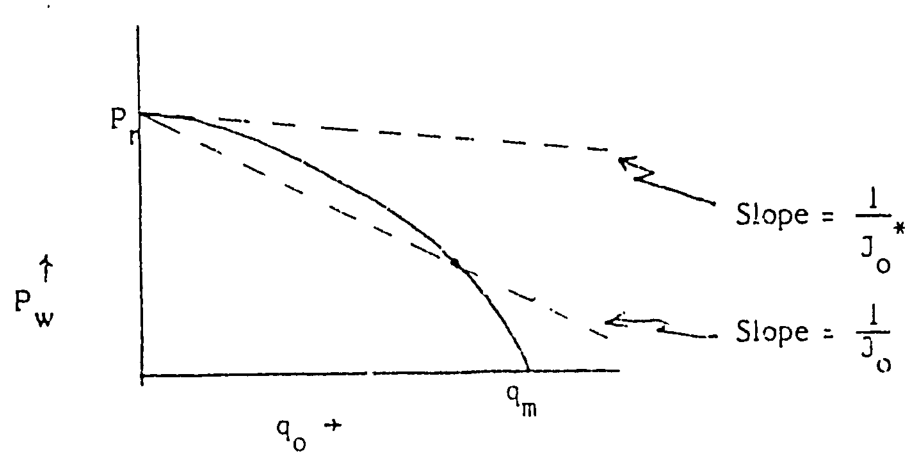
can be factored and rearranged as

$$J_o = \frac{q_o}{P_r - P_w} = \frac{q_m}{P_r}\left(1 + .8 \frac{P_w}{P_r}\right)$$

and then one can define

$$J_o^* = \lim_{P_w \to P_r} J_o = 1.8 \frac{q_m}{P_r}$$

This is the Oil Productivity Index at zero drawdown.



From Darcy's law (production q_o here positive)

$$q_o = \frac{2\pi rh}{B_o}\ \frac{K_o}{\mu_o}\ \frac{\partial P}{\partial r}$$

so

$$\int_{r_w}^{r_e} q_o \frac{dr}{r} = 2\pi h \int_{P_w}^{P_e} \frac{K_o}{\mu_o B_o}\ dP$$

Now as $P_w \rightarrow P_r$ <u>all</u> pressures will approach P_r and saturation distribution can rearrange to uniform S_o, then $q_o \rightarrow o$ everywhere but $K_o/\mu_o B_o \rightarrow$ const.; obtain:

$$J_o^* = \lim_{P_r \rightarrow P_w} \mathcal{J}_o = \frac{2\pi h}{\ln \frac{r_e}{r_w}} \left(\frac{K_o}{\mu_o B_o}\right)$$

Thus J_o^* is proportional to $K_o/\mu_o B_o$ at the reservoir pressure P_r.

As reservoir is produced P_r changes and $K_o/\mu_o B_o$ changes because both P_r and S_o are changing.

*Use material balance to forecast new P_r and S_o, compute new values of K_o, μ_o, B_o, say P_r', μ_o', B_o'. Thus

$$\frac{J_o'^*}{J_o^*} = \frac{\left(\frac{K_o'}{\mu_o' B_o'}\right)}{\left(\frac{K_o}{\mu_o B_o}\right)} = \frac{\frac{q_m'}{P_r'}}{\frac{q_m}{P_r}}$$

*Thus if from well test data, P_r, q_o, P_w a q_m is computed from Vogel Equation and also K_o, μ_o, B_o are determined, one can forecast the <u>new</u> value of q_m, q_m', at a future state of the reservoir.

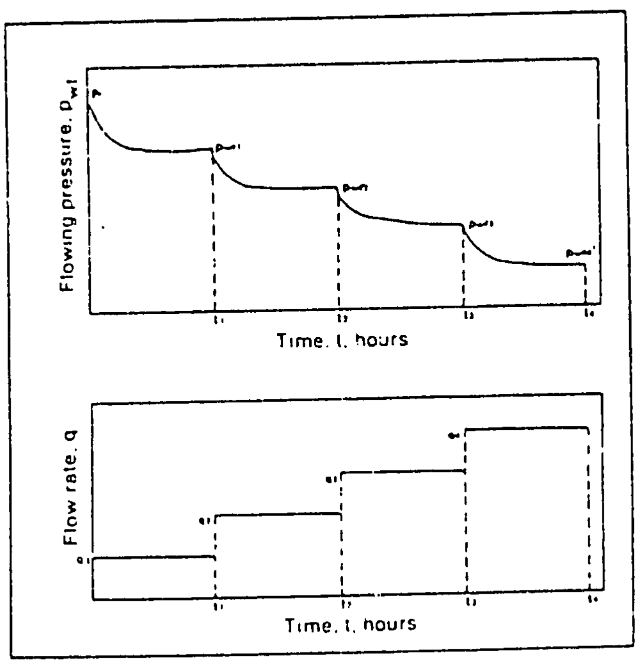
$$*q_m' = \frac{P_r'}{P_r}\ q_m\ \frac{K_o'\mu_o B_o}{K_o \mu_o' B_o'}$$

*Then, with q_m' and P_r' one can forecast a future value of q_o for <u>any</u> P_w using Vogel Eq.

$$q_o = q_m' \left(1 - .2 \left(\frac{P_w}{P_r'} \right) - .8 \left(\frac{P_w}{P_r'} \right)^2 \right)$$

Flow-After-Flow Tests; Gas or Oil

When a shut-in well, either gas or oil, is opened to flow the bottom-hole pressure declines as the pressure distribution in the reservoir changes with time. With the flow rate restricted through an orifice choke, or partially open valve, the flow rate and bottom-hole pressure will stabilize to essentially constant values. The valve may then be opened further resulting in another transient period until a new q and P_w are established. Typical data for this flow-after-flow test are shown schematically below.



For such pseudo-steady-state conditions as indicated by the pairs of values P_{w_1}, q_1 ; P_{w_2}, q_2 ; etc. it can be shown from Darcy's law that one should find for a gas well

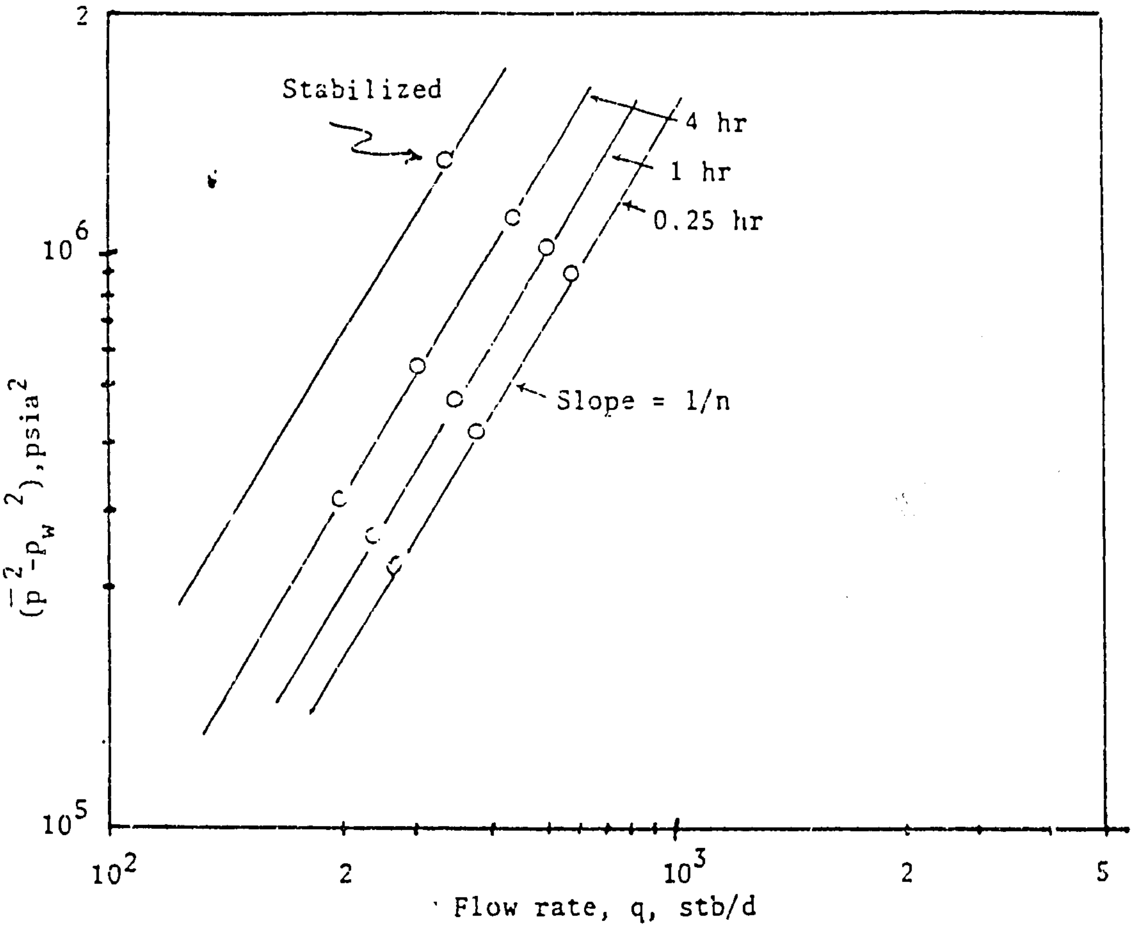
$$\frac{P_r^{\,2} - P_{wi}^{\,2}}{q_i} \approx \text{constant}$$

and the value of this constant characterizes the well production capability. In fact

$$J' \approx \frac{2q_i P_r}{P_r^2 - P_{wi}^2} = \frac{q_i}{P_r - P_{wi}} \left(\frac{P_r}{\bar{P}_i} \right)$$

is essentially the same quantity as the productivity index of an equivalent oil well. Here $\bar{P}$ is $(P_r + P_{wi})/2$. Conventionally one plots on log-log paper $P_r^2 - P_{wi}^2$ vs q_i for the gas well and draws the best straight line.

The difficulty with the flow-after-flow test is the long times required to reach stabilized flow conditions. A modified procedure, also used in oil wells, has been developed to circumvent this difficulty, it is called an <u>isochronal flow test</u>. The well is shut-in then opened to flow at a fixed rate for a period of time Δt, with P_w read at times Δt_1, Δt_2, Δt_3, etc. The well is shut in again for a time equal to the total flow time. The process is repeated for a higher rate, then for another, etc. Finally at the last rate the well is allowed to flow until a stabilized P_w is reached. From these data on the gas well one can plot $P_r^2 - P_w^2$ vs q <u>for equal times of flow</u>. i.e., using the P_w say at 30 minutes of flow for each of the flow rates. On log-log paper this should be a straight line whose slope is approximately unity for a gas well. This is done for each of the elapsed times. Finally the <u>one</u> data point for stabilized flow is plotted and a parallel line drawn as shown in the sketch below.



The point of this test is to establish a prediction equation for stabilized flow conditions, thus defining a J' and an n for the equation (Fetkovitch)

$$q = J' (P_r^2 - P_w^2)^n$$

Normally one would require at least two points (q, P_w) at stabilized conditions, which in a tight gas formation could require days to establish. This method seems to effectively avoid the need for more than one stabilized flow point because the above equation seems to fit data at corresponding times with the same value of n.

Transient Pressure Tests

The basis for transient well testing resides in the fact that for single-phase flow, at pressures above the bubble point, Darcy's law and a material balance on fluid mass yield the equation

$$\nabla \cdot \left(\frac{k}{\mu} \nabla (P - \rho g z) \right) = \phi c \frac{\partial P}{\partial t}$$

governing the transient pressure history within the reservoir. Here ρ is the approximately <u>constant</u> fluid density, μ its viscosity, c the effective compressibility of the fluid-rock system, and k and ϕ the permeability and porosity of the reservoir. Treating k as uniform this approximates further to

$$\nabla^2 P = \frac{\phi \mu c}{k} \frac{\partial P}{\partial t}$$

for P, or for $P - \rho g z$ with $P - \rho g z$ replacing P. This is the same form as the equation for diffusion or heat conduction.

The solution for a well of zero radius penetrating a reservoir of thickness h producing at constant rate q from an infinite reservoir with static pressure P_r is

$$P = P_r + \frac{70.6\, q\, \mu\, B}{kh} \; Ei \left(- \frac{\phi \mu c r^2}{632.\, kt} \right)$$

where

P_r = shut-in pressure (Psi)

P = pressure at r, t (Psi)

q = flow rate (STB/D)

h = thickness of formation (ft)

c = effective compressibility (Psi^{-1})

ϕ = porosity (fraction)

r = radial distance from well (ft.)

t = time on production (days)

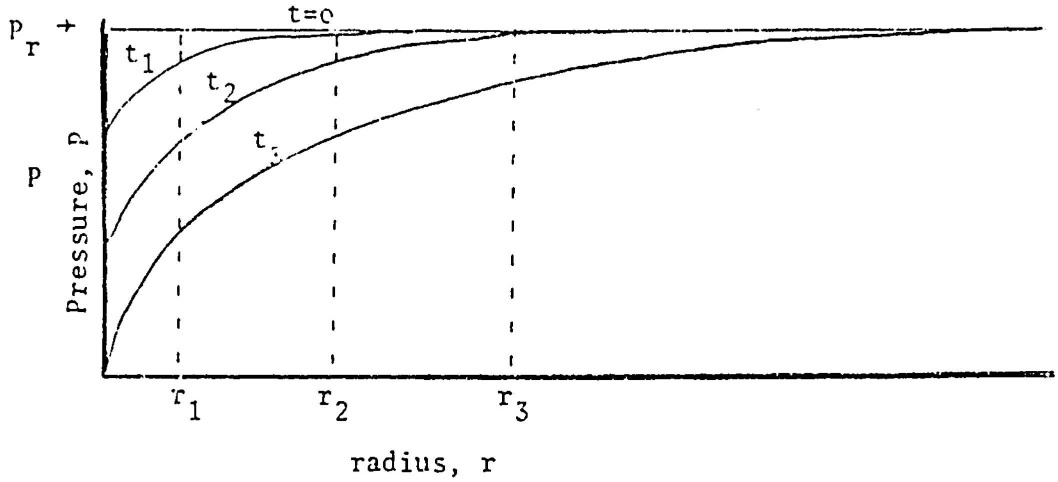
B = Formation volume Factor (Res BBl/STB)

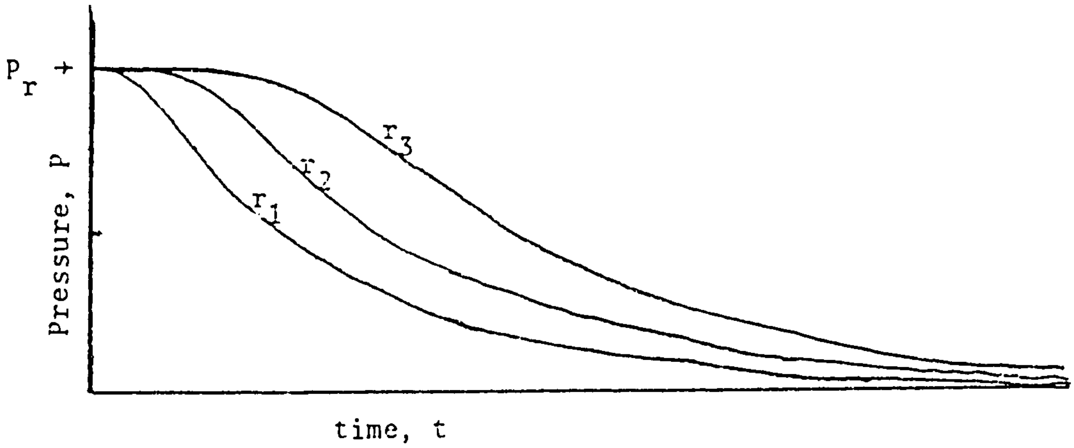
Here Ei is a tabulated function defined by

$$- Ei(-x) = \int_{x}^{\infty} \frac{e^{-\xi}}{\xi} d\xi$$

and called the Exponential Integral Function. For $x < 0.5$ the approximation $- Ei(-x) \approx - .5772 - \ell n (x)$ holds very well.

The general behavior of this solution is shown by the following sketches which provide a lot of intuitive insight into flow and pressure behavior of wells.





Note that in view of the approximation above we have for <u>large t</u>, or <u>small r</u>,

$$P \approx P_r \frac{-70.6\ q\mu B}{kh} \left[-2\ \ell n\ r + n\ \frac{6.32\ kt}{\phi \mu c} + .5772 \right]$$

This is called the pseudo-steady-state equation because for an incompressible fluid ($c = 0$) and a well flowing at constant rate q the differential equation is

$$\nabla^2 P = 0$$

and the solution is

$$P = \text{constant} + \frac{70.6\, q \mu B}{kh} \ln r$$

with the constant arbitrarily determined by fixing P at some r. This is truly a steady-state solution.

Superposition Principle; Shut-in Pressure Build-Up Test

Since the differential equation above is linear solutions can be added to yield new solutions thus, for example, to represent the pressure history in a well of bore radius r_w with the rate history

$$q = q \quad , \quad 0 < t < t_s$$
$$q = 0 \quad , \quad t > t_s$$

we have approximately for small r_w

$$P = P_r - \frac{70.6\, q \mu B}{kh} \left[\ln \frac{6.32\, kt}{\phi \mu c r_w^2} + .5772 \right] \quad ,$$

$$\text{for} \quad 0 < t < t_s$$

and

$$P = P_r - \frac{70.6\, q \mu B}{kh} \left[\ln \frac{6.32\, kt}{\phi \mu c r_w^2} + .5772 \right] \quad ,$$

$$+ \frac{70.6\, q \mu B}{kh} \left[\ln \frac{6.32\, k(t-t_s)}{\phi \mu c r_w^2} + .5772 \right]$$

$$\text{for} \quad t > t_s$$

This is a superposition, for $t > t_s$, of a well of production rate q started at time zero and a well of production rate $-q$ started at time t_s. This simplifies to

$$P = P_r - \frac{70.6\, q \mu B}{kh} \ln \frac{t_s + \Delta t}{\Delta t}$$

with Δt being elapsed shut-in time after producing at rate q for time t_s.

Thus if bottom-hole pressure is recorded as a function of time following shut-in a plot of P versus $(t_s + \Delta t)/\Delta t$ on semi-log paper should approach a straight line whose slope gives a value of kh.

Factors Affecting Build-up Tests

Obviously the above analysis is a gross over simplification of the physical situation in a pressure build-up test. Some factors not accounted for in the simple analysis are:

(1) well-bore size

(2) reservoir heterogeneity

(3) reservoir boundaries

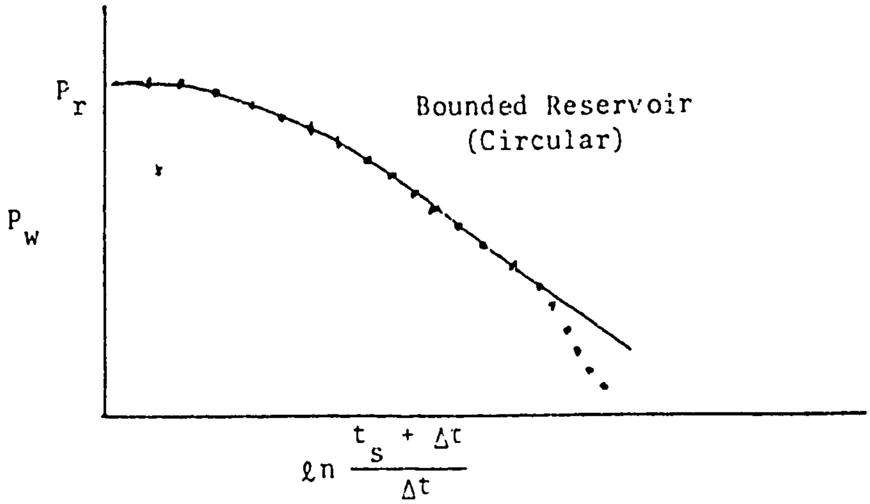
(4) interference from other wells

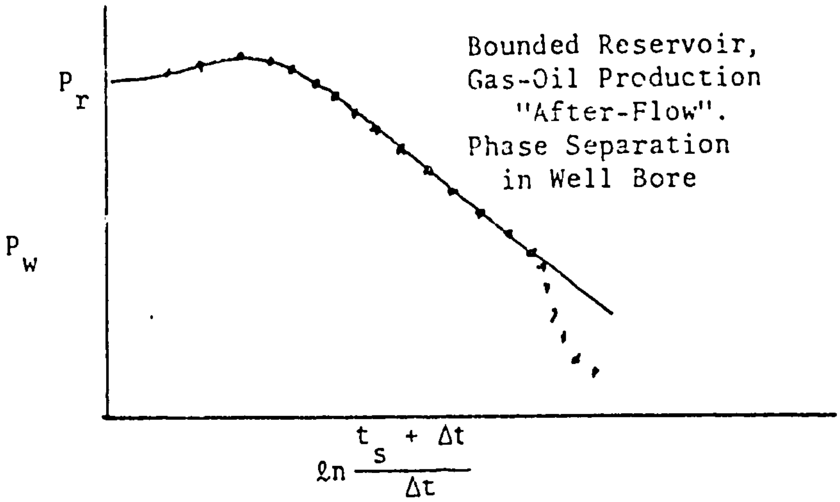
(5) multi-phase flow

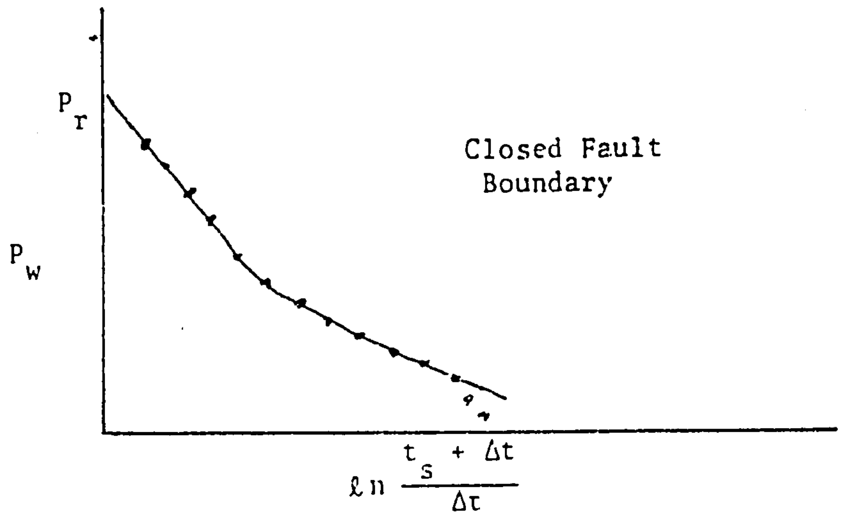
(6) variable rate history

(7) after-flow, q not zero at bottom-hole

Many of these factors have been successfully treated and the literature on the subject is voluminous. Sketches below indicate effects due to some of these factors.


P_r
P_w
Bounded Reservoir
(Circular)
$\ell n \frac{t_s + \Delta t}{\Delta t}$


P_r
P_w
Bounded Reservoir,
Gas-Oil Production
"After-Flow".
Phase Separation
in Well Bore
$\ell n \frac{t_s + \Delta t}{\Delta t}$


P_r
P_w
Closed Fault
Boundary
$\ell n \frac{t_s + \Delta t}{\Delta t}$

Other Transient Pressure Tests

A wide variety of transient tests and methods of interpretation have been devised. "Pulse testing" between wells is used to determine permeability and porosity in the region between wells. This involves "pulsing" one well by producing for a brief period then shutting in and detecting the pressure transient in an adjacent well. Vertical permeability tests based upon early pressure transients in partially penetrating wells, or injection at one point in a well and producing at a neighboring point above or below in the same well, have been proposed.

Perhaps the most useful modification of the simple test theory has been in the incorporation of well bore damage effects . This is described as follows.

Well Damage Evaluation: "Skin Effect"

Van Everdingen (1953) introduced the notion of "skin effect" to characterize the effect of near well-bore permeability modification on well-bore pressure transients. This can be described simply by asserting that in addition to the draw-down

$$P_r - P_w \approx - \frac{70.6\, B\, q\, \mu}{kh} \quad Ei\left(- \frac{\phi \mu c r_w^2}{6.32\, kt} \right)$$

which exists due to pressure losses in the clean formation, arising from flow rate q, there is an additional draw-down required to move fluid through a well bore "skin" at the rate q, this being given by

$$\Delta P_s = + \frac{70.6\, B\, q\, \mu}{kh} \quad S_D$$

with S_D a dimensionless factor determined by the "skin". Thus adding this "correction"

$$P_r - P_w = -\frac{70.6\ B\ q\mu}{kh}\left[-S_D + Ei\left(\frac{\phi\mu c r_w^2}{6.32\ kt} \right) \right]$$

or, approximately, for large flowing time t

$$P_w \approx P_r - \frac{70.6\ B\ q\mu}{kh}\left[S_D + \ell n \frac{6.32\ kt}{\phi\mu c r_w^2} + .5772 \right]$$

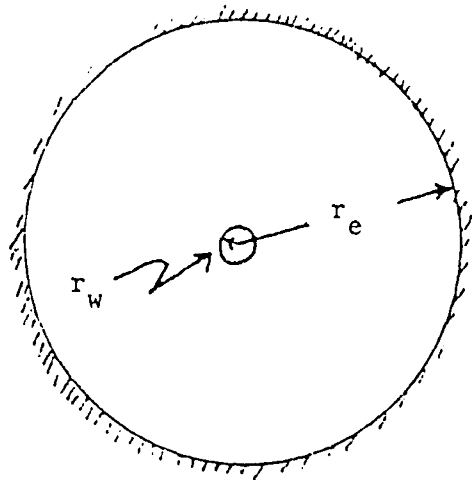
S_D is called the "skin factor" of the well.

The analysis of shut-in pressure build-up is not changed by this, thus plotting shut-in P_w vs $\ell n\,(t_s + \Delta t)/\Delta t$ still gives kh. Hence if ϕ and c are known one uses this equation on flowing data to estimate S_D.

ν\

Bounded "Circular" Reservoir

Consider a bounded, uniform, circular reservoir,



For flow of slightly compressible liquid (above bubble point), the equation of state is

(1) $$\rho = \rho_o e^{c(P-P_o)}$$

or for small compressibility, c,

(2) $$\rho \approx \rho_o\,(1 + c\,(P-P_o))$$

where c is actually defined by

(3) $$c = \frac{1}{\rho}\frac{\partial \rho}{\partial P}$$

as the compressibility with ρ mass density.

Conservation of mass requires

(4) $$-\nabla \cdot (\rho\,\hat{v}) = \phi\,\frac{\partial \rho}{\partial t}$$

and with Darcy's law

(5) $$\hat{v} = -\frac{k}{\mu}\,(\nabla P - \rho\hat{g})$$

this gives for horizontal flow and small c

(6) $$\frac{k}{\mu}\,\frac{1}{r}\,\frac{\partial}{\partial r}\left(r\,\frac{\partial P}{\partial r}\right) = \phi c\,\frac{\partial P}{\partial t}$$

in the radial geometry. In this we can replace P by $\psi = P - \rho_0 g z$ with ρ_0 constant as an approximation. i.e., the P determined as solution of (6) is in <u>one</u> plane at $z = 0$.

<u>Boundary Value Problem</u>

$$(7)\quad \begin{cases} \dfrac{1}{r}\dfrac{\partial}{\partial r}\left(r\dfrac{\partial P}{\partial r}\right) = \dfrac{\phi\mu c}{k}\dfrac{\partial P}{\partial t} \\ \dfrac{\partial P}{\partial r} = 0 \quad , r = R \\ -2\pi r_w h \dfrac{k}{\mu}\dfrac{\partial P}{\partial r} = -qB_o \quad , r = r_w \\ P = P_o \quad , t = 0 \end{cases}$$

separate variables, obtain

$$(8)\quad P - P_o = \sum_\beta \left(A_\beta J_o(\beta r) + B_\beta Y_o(\beta r)\right) e^{-\frac{kt}{\phi\mu c}\beta^2} + G(r,t)$$

where $G(r,t)$ is any special solution of (7) we may need. For boundary condition at $\underline{r = R}$ this then becomes

$$(9)\quad P - P_o = \sum_\beta \bar{A}\left(Y_o'(\beta R) J_o(\beta r) - J_o'(\beta R) Y_o(\beta r)\right) e^{-\frac{kt}{\phi\mu c}\beta^2} + G(r,t)$$

Provided that $G(r,t)$ also satisfies the condition at R

$$(10)\quad \frac{\partial G}{\partial r}(R,t) = 0$$

For the condition at $\underline{r = r_w}$ we then need

$$(11)\quad Y_o'(\beta R) J_o'(\beta r_w) - J_o'(\beta R) Y_o'(\beta r_w) = 0$$

and

$$(12)\quad \frac{\partial G}{\partial r}(r_w,t) = +\frac{q\mu B_o}{2\pi k h r_w}$$

Eq. (11) provides an infinite set of roots β_j, $j = 1,2, \ldots$, and it is clear that the special solution $G(r,t)$ is needed !

Now show that usual zero separation constant solution is not acceptable for $G(r,t)$. For $\beta = 0$ the solution by separation of variables is

$$(13) \quad G = A \ln r + B$$

and while this can satisfy the requirement at $r = r_w$ it does not satisfy the requirement $\partial G/\partial r = 0$ at $r = R$. Thus this is not acceptable.

The $\beta = 0$ solution also corresponds to $\partial P/\partial t = 0$ and the next simplest special solution would be for

$$(14) \quad \frac{\partial P}{\partial t} = B = \text{constant}$$

Thus try this ! This in the differential equation gives:

$$(15) \quad \frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial P}{\partial r}\right) = \frac{\phi\mu c}{k} B$$

which integrates to give

$$(16) \quad G = P - P_o = \frac{\phi\mu c}{k} B \frac{r^2}{4} + A \ln r + C + Bt$$

as the special solution. Here A and C are constants of integration. Since C is still arbitrary modify C to make the argument of $\ln$, $\ln r$, into proper dimensionless form, thus

$$(17) \quad G = \frac{\phi\mu c}{k} B \frac{r^2}{4} + A \ln \frac{r}{R} + \bar{C} + Bt$$

is the special solution. Fixing $\partial G/\partial r = 0$ at $r = R$ gives

$$(18) \quad A = -\frac{\phi\mu c}{k} B \frac{R^2}{2}$$

Then the condition at $r = r_w$ in (12) gives with,

$$(19) \quad G = \frac{\phi\mu c}{k} \frac{B}{2} \left[\frac{r^2}{2} - R^2 \ln \frac{r}{r}\right] + \bar{C} + Bt$$

24

the result

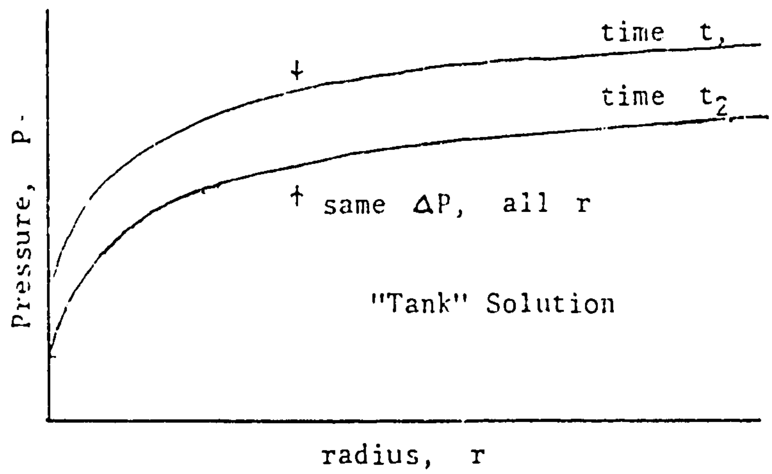
$$(20) \qquad B = -\frac{qB_o}{\pi[R^2-r_w^{\ 2}]h\phi c}$$

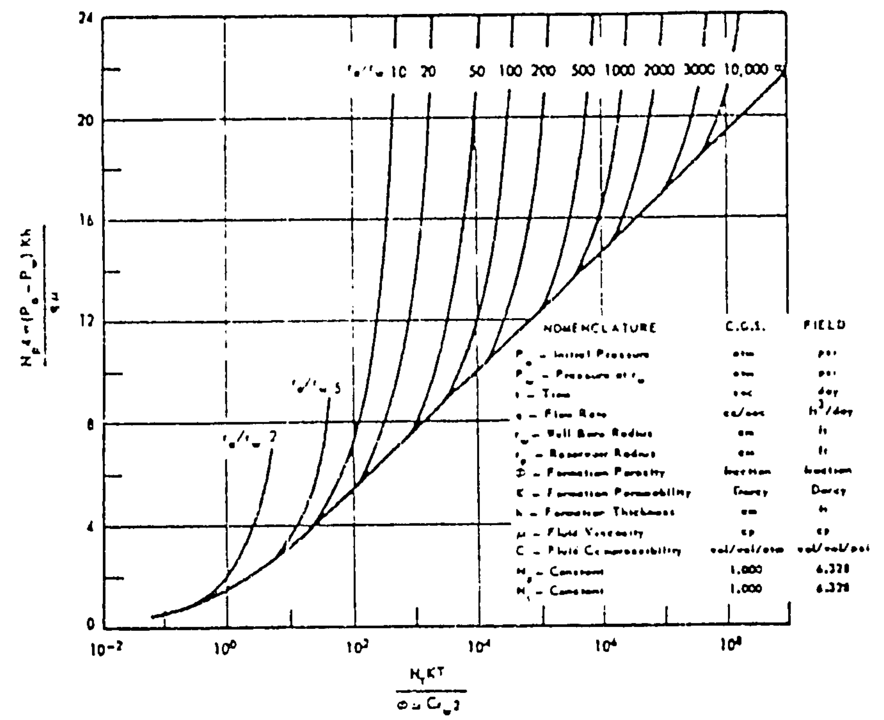
and if we arbitrarily set $\bar{C}$ to zero we have the solution

$$(21) \quad P - P_o = \sum_{j=1}^{\infty} \bar{A}_j \left[Y'_o(\beta_j R)\, J_o(\beta_j r) - J'_o(\beta_j R)\, Y_o(\beta_j r) \right] e^{-\frac{kt}{\phi\mu c}\beta_j^2}$$
$$- \left\{ \frac{qt\,B_o}{\pi[R^2-r_w^2]\,h\phi c} + \frac{q\mu\,B_o}{2\pi kh\,[R^2-r_w^2]} \left[\frac{r^2}{2} - R^2 \ln \frac{r}{R}\right] \right\}$$

which satisfies all conditions except at $t = 0$. Finally then setting $t = 0$ and using the orthogonality of the linear combination of Bessel functions we obtain the $\bar{A}_j$ and solution is complete. Thus we have justified setting $\bar{C}$ to zero. Hurst and Van Everdingen (1949) obtained this solution in a different way and give the evaluation of the $\bar{A}_j$.

The special solution $G(r,t)$ is the asymptotic solution for $P - P_o$ as $t \to \infty$. Note that in the series all exponentials go to zero. This is a pseudo steady-state solution with the same distribution of $P - P_o$ versus r at any time but shifted up or down depending on sign of q. q is positive for production, negative for injection. We call this the "Tank Solution". The sketch below shows the general behavior of this solution.





Compressible liquid flow; flowing pressure in a well at the center of a circular reservoir. (*After Hurst and Van Everdingen, 1949.*)

26

Drill Stem Testing

Well testing as already described applies to completed wells but flow tests, pressure tests and fluid sampling are also carried out on wells before the well is completed. Such tests are useful to evaluate a zone in a well for completion.

Method

Drill stem testing is carried out with tool mounted on end of drill pipe string. It consists of:

- packer
- flow control valves
- pressure gauge
- Fluid sampler

Different service companies offer various designs but basic elements are the same.

There is also a wire-line tool by Schlumberger that functions with the same elements but two fluid collection chambers.

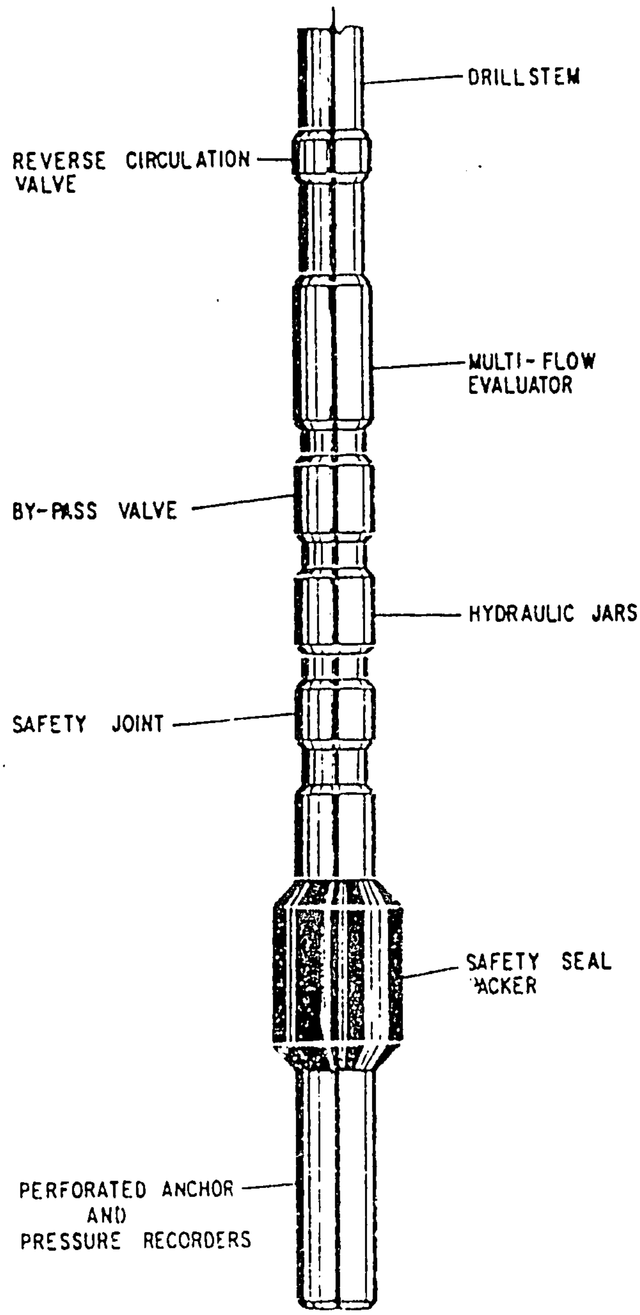
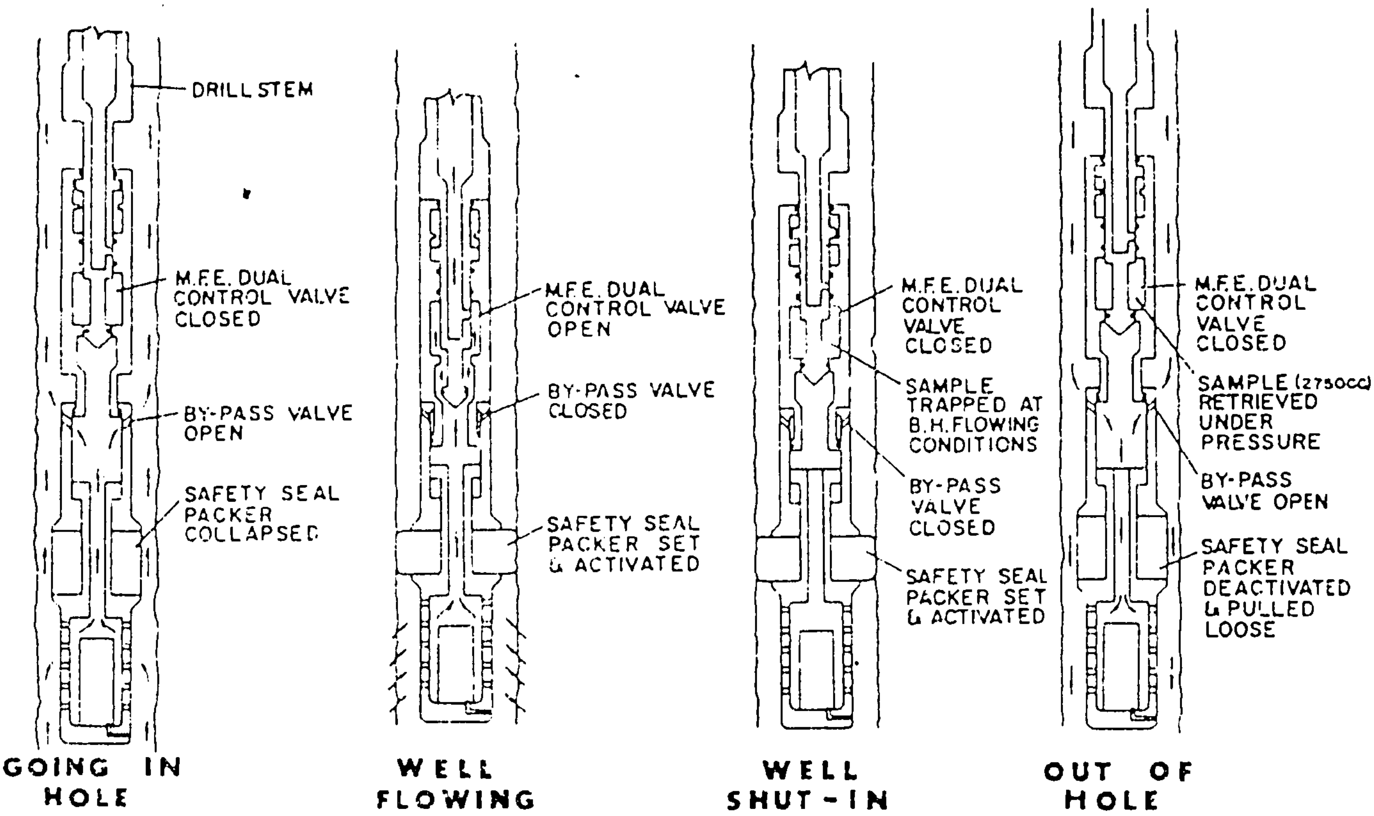


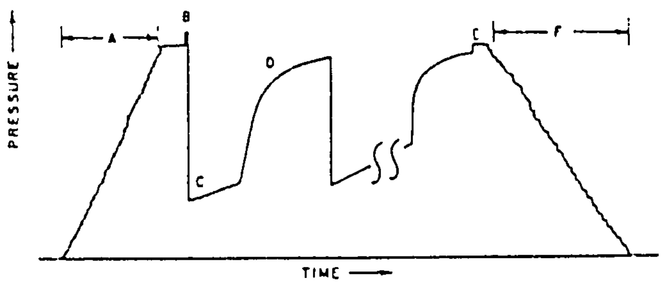
Diagram of currently operational DST tool. (After McAlister, Nutter and Lebourg.)

21



Sequence of operations for MFE tool. (After McAlister, Nutter and Lebourg.[3])

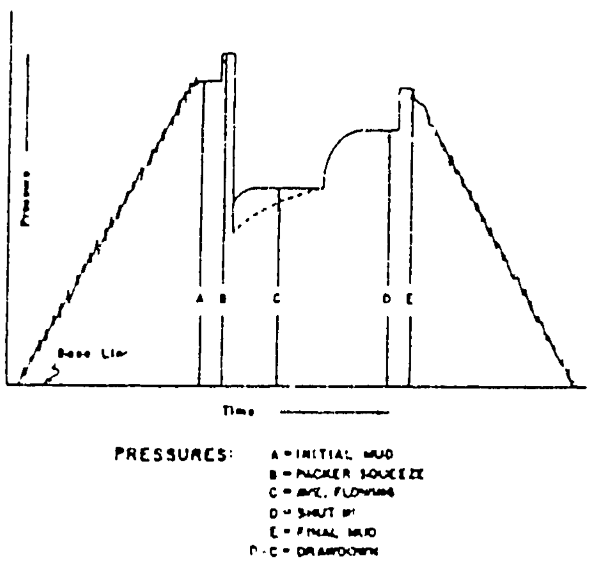
Above is sequence of operations. At right is typical pressure record on Amerada type pressure bomb in tool



Schematic DST pressure record.

A. going in hole
B. Packer set
C. (to break) flowing
D. Shut-in pressure build-up. Between D-E a second flow-shut-in.
E. Unseat packer
F. Coming out of hole

NOTE: First flow removes mud filter cake and some well damage. Second flow period is more representative of formation.



TYPICAL CHARTS FROM A SATISFACTORY TEST

Pressure Build-up Analysis

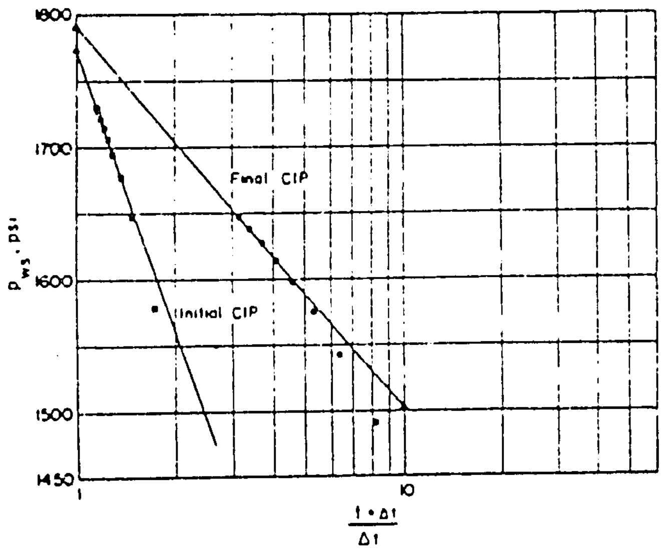
Fluid flow is through a choke (orifice) and if critical flow occurs then flow rate in flowing period is essentially constant. If this occurs then conventional pressure build-up analysis applies to shut-in pressure data.

Shown here is conventional "Horner" plot of shut-in pressure, P_{ws}, versus

$$\ell n \frac{t_F + \Delta t}{\Delta t}$$

with t_F flow time, Δt shut-in time.

$$\text{Slope} = \frac{-70.6\, q\mu B}{kh}$$

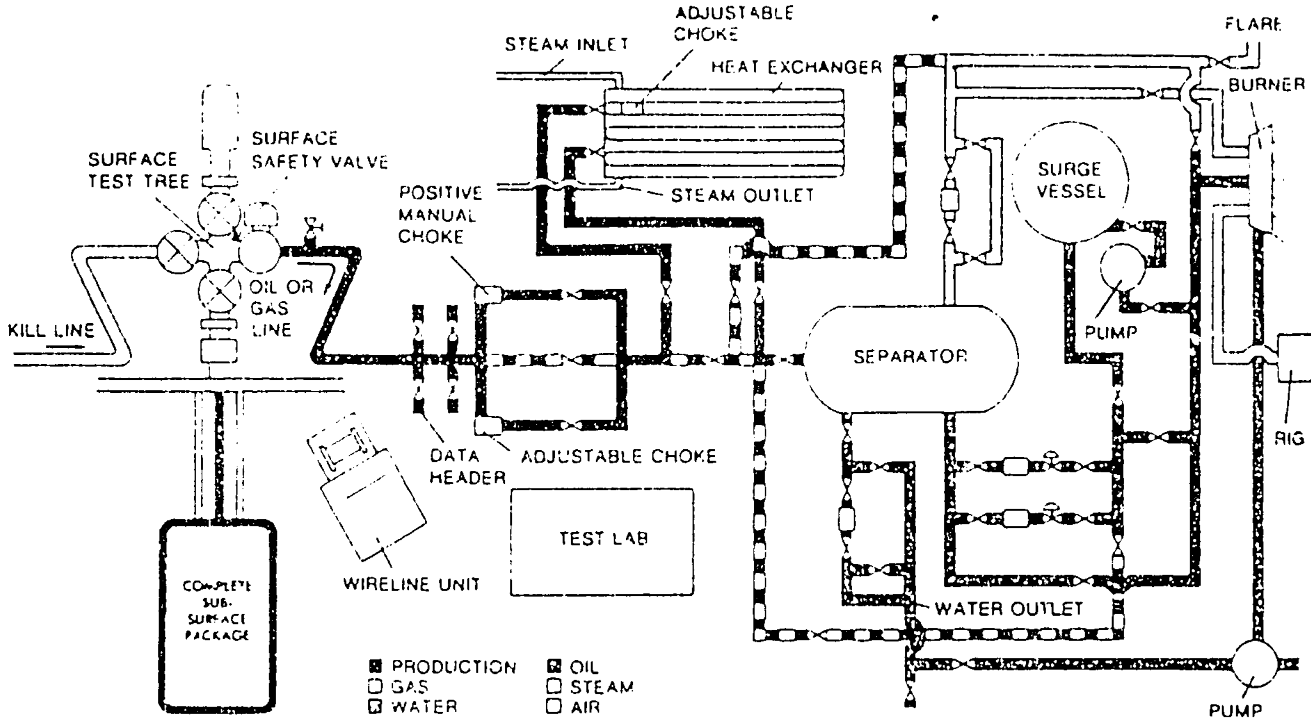


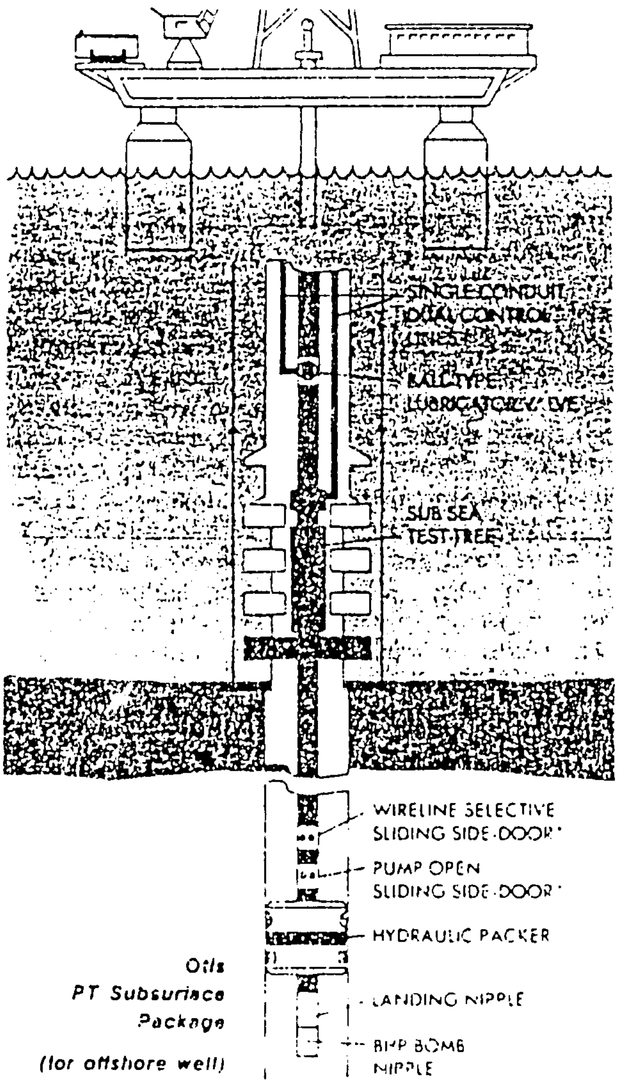
Field example of DST pressure buildup curves. (After Maier.[1])

NOTE: "Final" shut-in gives greater kh than "Initial" shut-in, indicating clean-up in first flow period. Also note that to estimate kh q must be estimated from collected fluid volume in drill pipe and a value for μ is also required.

Fluid Sampler

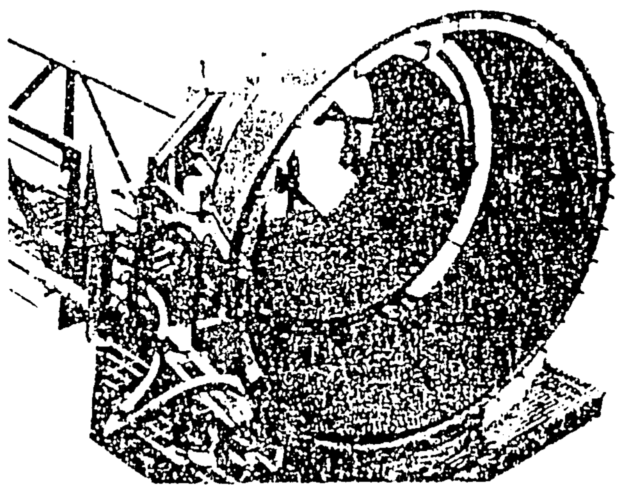
Note that the above tool collects a fluid sample at bottom-hole pressure and temperature.





Otis PT Subsurface Package (for offshore well)

When properly installed, an Otis Sub Sea Test Tree is designed to allow standard well productivity tests to be conducted from a floating vessel without removing the blow-out preventer stack. The tree is designed to be released hydraulically, pneumatically and, if necessary mechanically should you need to abandon the well. Relatching is accomplished by following relatively simple reengaging procedures.



OTIS/NAO CB-12 WELL TEST BURNER

This Otis Service is designed to provide detailed information on well production such as:

1. Deliverability, establishing productivity rates and temperatures at wellhead.
2. Flow capacity of reservoir section penetrated by the well.
3. Establish a pressure drawdown figure on the well (difference between shut-in and flowing pressure).
4. Confirmed deliverability and stabilized flow rate once rate of pressure drawdown becomes constant over a period of time.
5. Well's productivity index, whether very low or very high.
6. Conditions at formation face.
7. Porosity and permeability of formation or amount of fluid formation might hold, plus ability to give up this fluid.
8. Amount of permeable/porous rock (surrounding the well bore and reservoir) containing hydrocarbons and capability of flowing this fluid.
9. Reservoir limits or total area of hydrocarbons trap.
10. Absolute Open Flow (A.O.F.) potential, if gas well.
11. Back-pressure plot on gas well's performance.

GAS LIFT

Although the majority of artifical lift wells utilize pumping techniques (85.3%) the majority of those which produce larger quantities of oil (non-stripper wells) utilize gas lift as a means of production (53%).

Gas lift involves the injection of high pressure gas (900-1500 psi) into the flowing stream with the objective of providing sufficient energy to lift the fluid to the surface. Two techniques are used: continuous gas injection and intermittent gas injection:

Continuous Injection: Gas is introduced continuously at a controlled rate with the objective of decreasing the gradient of the mixture (oil + water + gas) flowing in the well.

Intermittent Injection: Gas is introduced periodically at high rate and for a short time (2-10 minutes) with the objective of lifting the fluid in the well by the rapid expansion of the injected gas slug.

The following table indicates general guidelines for application of the two systems:

Continuous: High/moderate P. I. wells with reasonable bottom hole pressure in relation to their depth.

Fluid production:

300 - 4000 B/D normal size tubing

4000 - 25000 B/D oversize tubing, annular flow

Intermittent: Low production rates either caused by low P. I. or low bottom hole pressure

Fluid production:

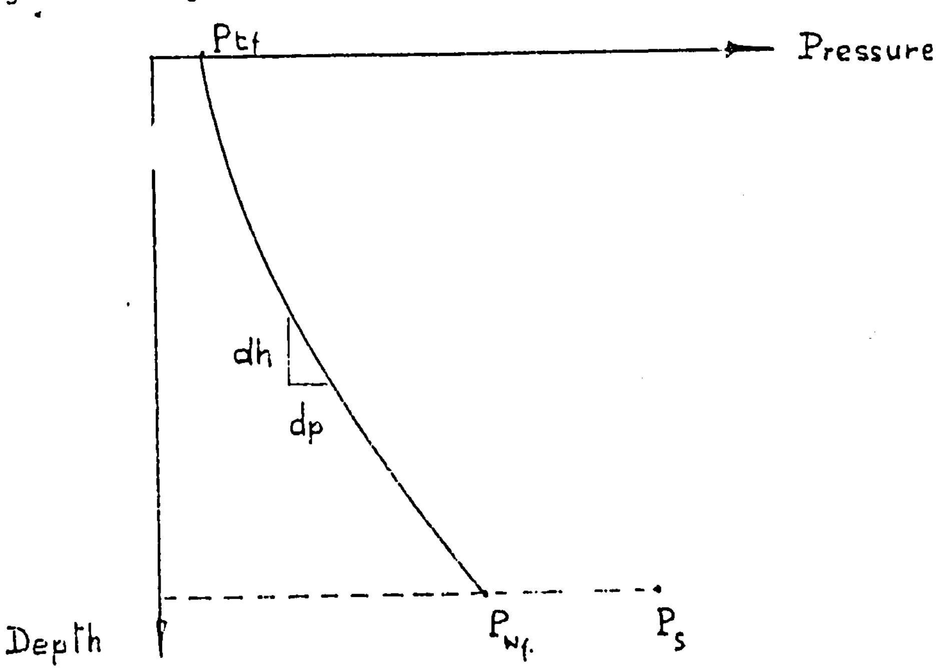
20 - 300 B/D normal tubing sizes

Two-Phase (Gas/Liquid Flow in Wells)

The following is a brief discussion of flow characterisitics in wells with the objective of establishing the principles used in gas lift design.

Pressure Traverse

Considering a well flowing at a steady liquid flow rate and gas/oil ratio. The following diagram represents the pressure-depths relation for a given tubing size. It is defined as a pressure traverse.



The inverse of the slope of the pressure traverse corresponds to the flowing pressure gradient (dP/dh). The non-linear character of the relation indicates that the gradient is a function of pressure. This is primarily due to the presence of the highly compressible gas phase.

At higher pressure (bottom of well) the actual volume of the gas is very small (even zero if pressure is above bubble point pressure). The gradient is mainly a function of liquid density and viscous losses (bubble type flow).

As pressure decreases the gas volume increases. Flow pattern changes to slug flow introducing different loss mechanisms (counter flow, slippage, momentum) and reducing mixture density.

At even lower pressure flow changes to annual flow, and in some cases to mist flow. Fluid velocity increases appreciably and frictional losses control the pressure gradient.

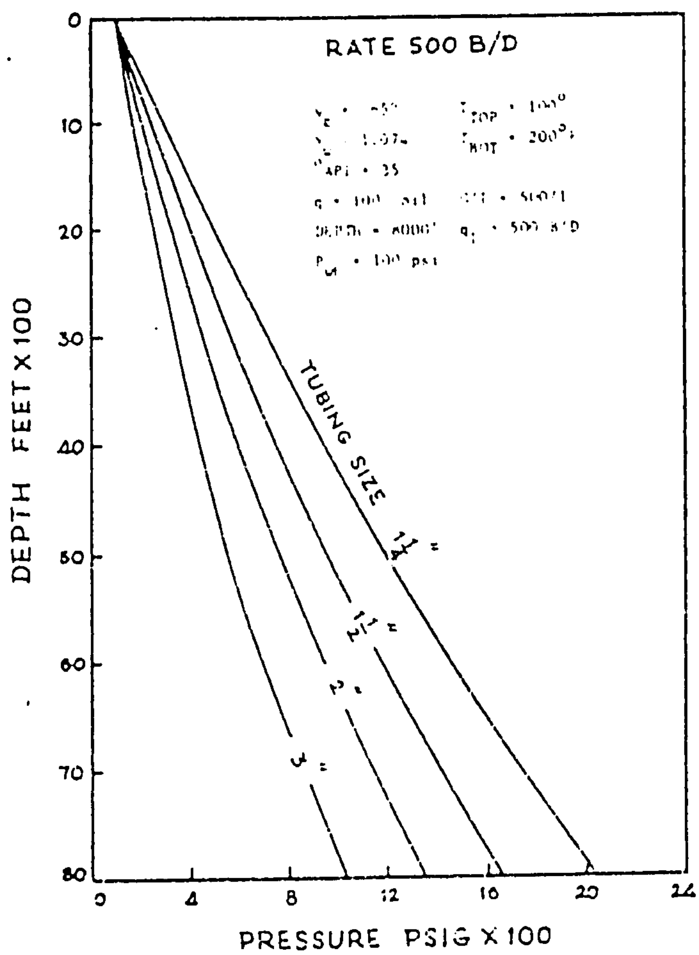
Mathematical description of the process relies on empirical correlations to predict gas/liquid distribution (flow pattern maps, liquid holdup) and energy losses. Discrepancies exist between calculation and observed results and between different methods of solution. A combination of the methods is generally used to cover the wide range of operating parameters.

Effect of Principal Variables

The flowing pressure traverse is principally controlled by: tubing size, gas-liquid ratio and liquid rate. Other variables include: flowing temperature, liquid density, water/oil ratio, fluid viscosity and surface tension.

Tubing Size

For a given liquid rate, gas-liquid ratio, surface pressure and fluid properties the following figure illustrates the effect of tubing size.



The flowing pressure at a given depth decreases as tubing size increases. (It should be noted however that for excessively large tubing in relation to the liquid and gas rates the flowing pressure increases due to gas slippage and accumulation of liquid in the well.)

The effect of tubing size for various liquid rates is shown in the following table.

TUBING SIZE EFFECT

(showing flowing bottom hole pressures)

	Rate			
Dia., in.	500	3,000	4,000	8,000
1¼	2,042	---	---	---
1½	1,680	---	---	---
2	1,371	---	---	---
3	1,042	1,592	1,819	---
4	---	1,319	1,459	2,068
5	---	1,025	1,072	1,285
6	---	950	972	1,092

TYPICAL WELL DATA

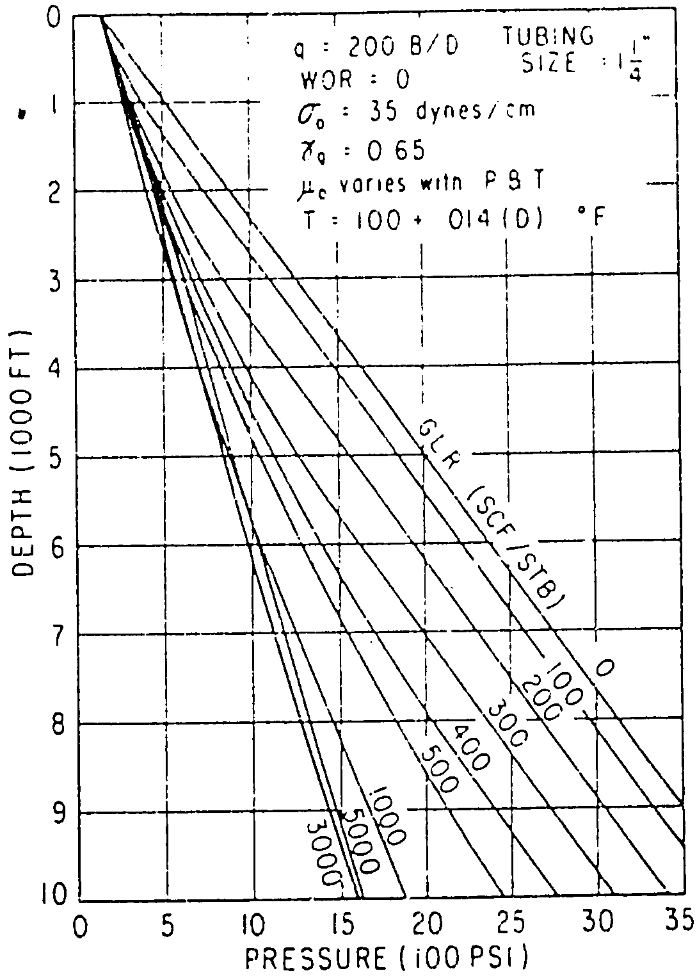
SGG	=	0.650		THT	=	100	°F
SGW	=	1.074		BHT	=	200	°F
SPI	=	35.000		DI	=	2.441	in.
CUT	=	0.000%		GLR	=	500	scf/stb
Depth	=	8,000	ft.	QL	=	500	stb
PWH	=	100	psig				

Gas-Liquid Ratio

For a given tubing size, liquid rate, surface pressure and fluid properties the following table and figure illustrate the effect of gas-liquid ratio.

GAS-LIQUID RATIO EFFECT

GLR	FBHP
0	2,938
100	2,669
200	2,234
300	1,783
400	1,398
500	1,175
600	1,042
800	913
1,000	862
1,500	801
3,000	752
5,000	768
10,000	915

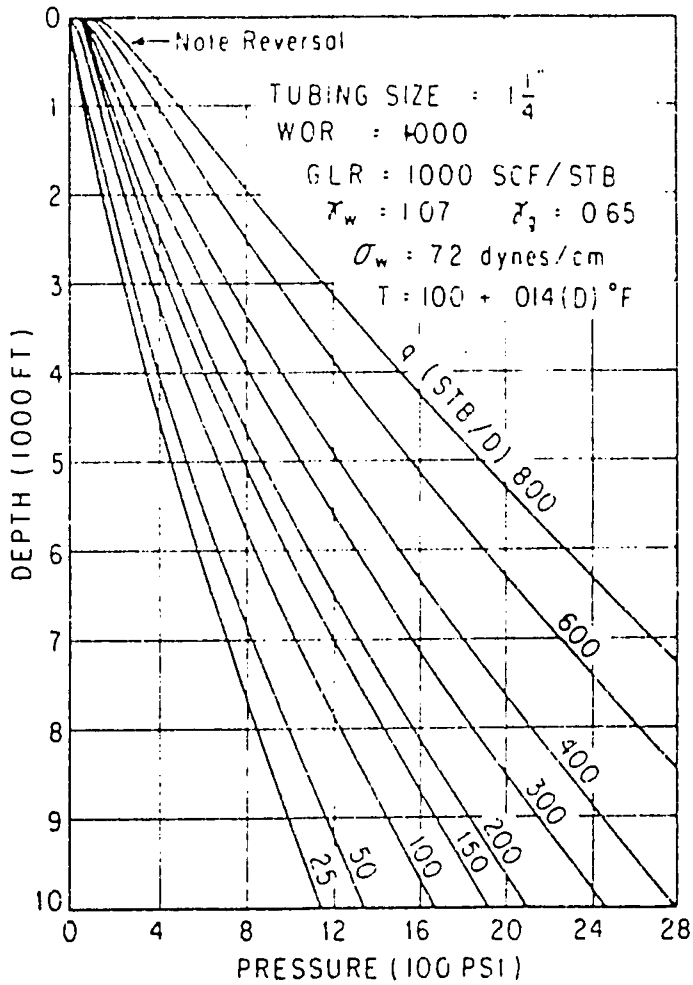


At a given depth the flowing pressure decreases as the gas-liquid ratio increases. The effect is reversed for gas-liquid ratios greater than a limiting GLR where the increase in mixture velocity causes increased frictional losses.

The limiting GLR corresponds to the minimum flow gradient that can be achieved in a given tub ize at a given liquid rate.

Liquid Rate

For a given tubing size, gas liquid ratio and fluid properties the following figure shows the effect of liquid rate.



The figure shows that at a given depth the flowing pressure increases as liquid rate increases. A marked change in pressure gradient occurs at the surface. This effect is due to the very low back pressure and the high mass rate in relation to the tubing size. It would not be observed in practical cases with properly sized tubing.

the pressure at some point in the well (usually BHP or well head pressure). The point on the chart at the same pressure for the particular gas-liquid ratio of the well, represents the point in the well. The chart can then be used to calculate the pressure-depth distribution assigning the correct depth to the known point and moving along the constant GLR curve up or down-hole as necessary.

Application of Continuous Gas-Lift

The production objective is generally expressed as a specific oil rate to be produced into a surface gathering system operating at a certain pressure. For a given well the oil rate is obtained by establishing a drawdown at the formation depending on productivity. The resulting flowing bottom hole pressure should be sufficient to move the fluid to the surface with enough pressure left at the tubing head to be able to flow into the surface systems.

If the formation pressure is insufficient the well may not flow or flow at a rate lower than the desired rate.

In solution gas expansion reservoirs formation pressure and productivity decline with increased cumulative recovery. Wells that may flow initially will stop flowing or flow at reduced rates. In water drive reservoirs the increased WOR requires increased total fluid production to maintain the desired oil rate. Also liquid density increases and gas-liquid ratio decreases as WOR increases.

Continuous gas lift is used to reduce pressure losses in the wellbore by introducing gas in the flowing stream thereby reducing the density of the fluid mixture.

General Guidelines

Gas Volume Requirements: 150-250 SCF of injected gas per barrel of fluid lifted per 1000 ft. of lift.

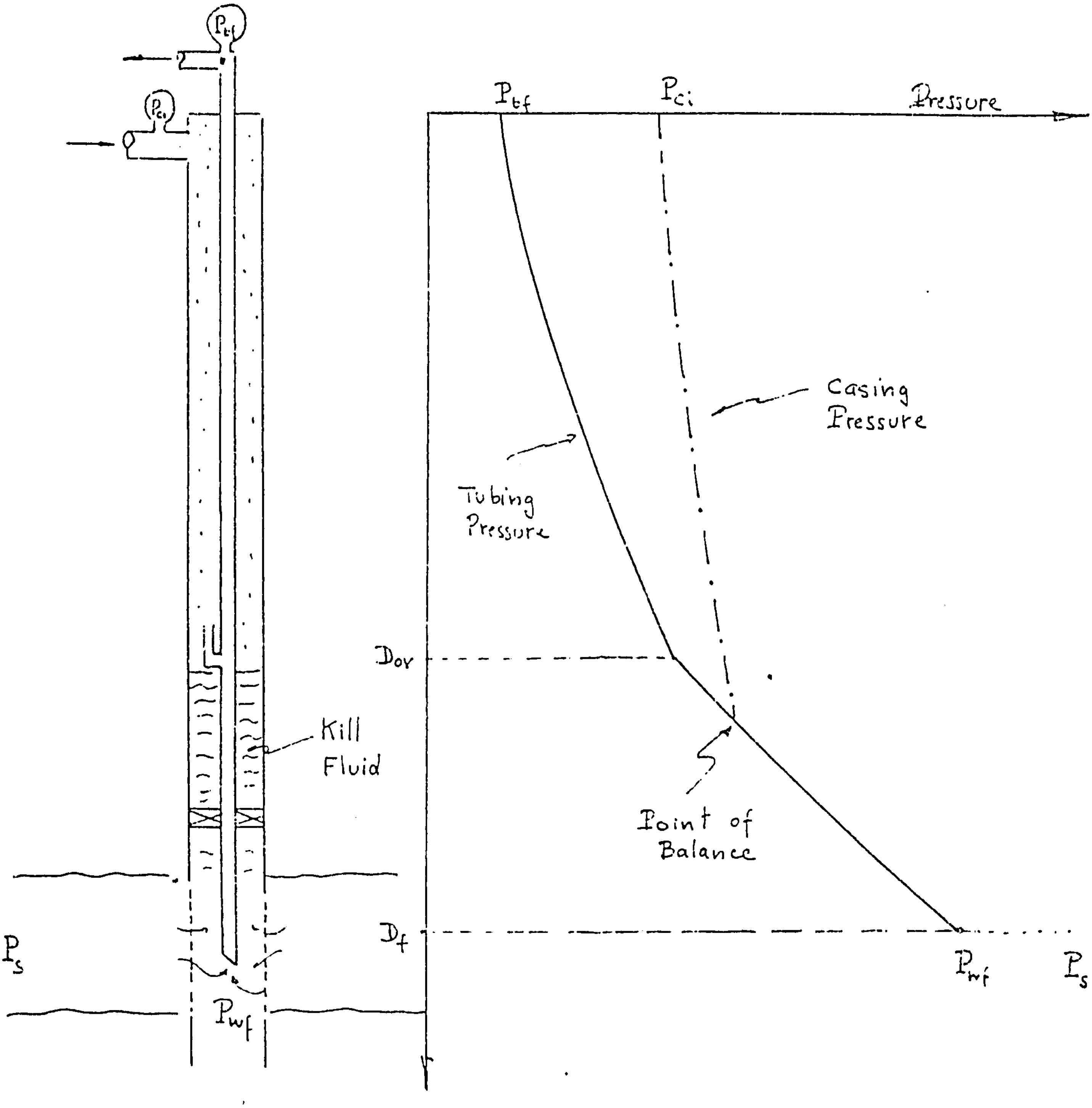
Injection Pressure Requirements: 100-150 psi per 1000 ft. of lift.

Maximum Depth of Lift: for normal tubing sizes <4" the depth of injection for given tubing (P_{tf}) and coring (P_{ci}) pressures is

$$\text{Depth} = \frac{P_{ci} - P_{tf}}{0.15}$$

Pressure Balance in Continuous Gas Lift

For steady state conditions of formation fluid production and gas injection the following diagram represents the pressure-depth relation existing in the well.



P_{tf}
P_{ci}
Kill Fluid
P_s
P_{wf}
P_{tf}
P_{ci}
Pressure
Casing Pressure
Tubing Pressure
D_{ov}
Point of Balance
D_f
P_{wf}
P_s

The point of balance corresponds to the depth at which casing pressure is equal to tubing pressure. In order to inject gas into the tubing the operating valve has to be above the point of balance. The distance from the point of balance depends on the pressure drop across the value seat due to flow of the required volume of injection gas. (ΔP_{valve} x 50-100 psi.)

The following relation can be established for average flowing gradients $\bar{\gamma}_{af}$ and $\bar{\gamma}_{bf}$ above and below the point of injection:

$$P_{tf} + \bar{\gamma}_{af} D_{ov} + \bar{\gamma}_{bf} (D_f - D_{ov}) = P_{wf}.$$

where

D_{ov} = depth of operating value

D_f = depth of formation

$\bar{\gamma}_{af}$ = gradient above injection is a function of volume of gas injected.

For a given flow rate P_{wf} is constant so that changes in D_{ov} and $\bar{\gamma}_{af}$ will result in changes in the flowing tubing pressure P_{tf}.

Gas Lift Design Problems

The obective is to design a system economically justifiable. Objective function expressed in terms of energy efficiency or in terms of present value when comparing alternative artificial lift systems.

The decision variables that are usually considered include:

Tubing Size

Flowline Size

Surface Gas Injection Pressure

Liquid Flowrate

Flowing Tubing Pressure

Injected Gas-Liquid Ratio

Separator Pressure

Two cases are relevant:

1. Flowing tubing pressure is independent of flow rate. This case involves situations where length of surface flowlines is small and separator pressure determines tubing head pressure.

2. Flowing tubing pressure is dependent on flowrate. This case involves significant length of surface flowlines.

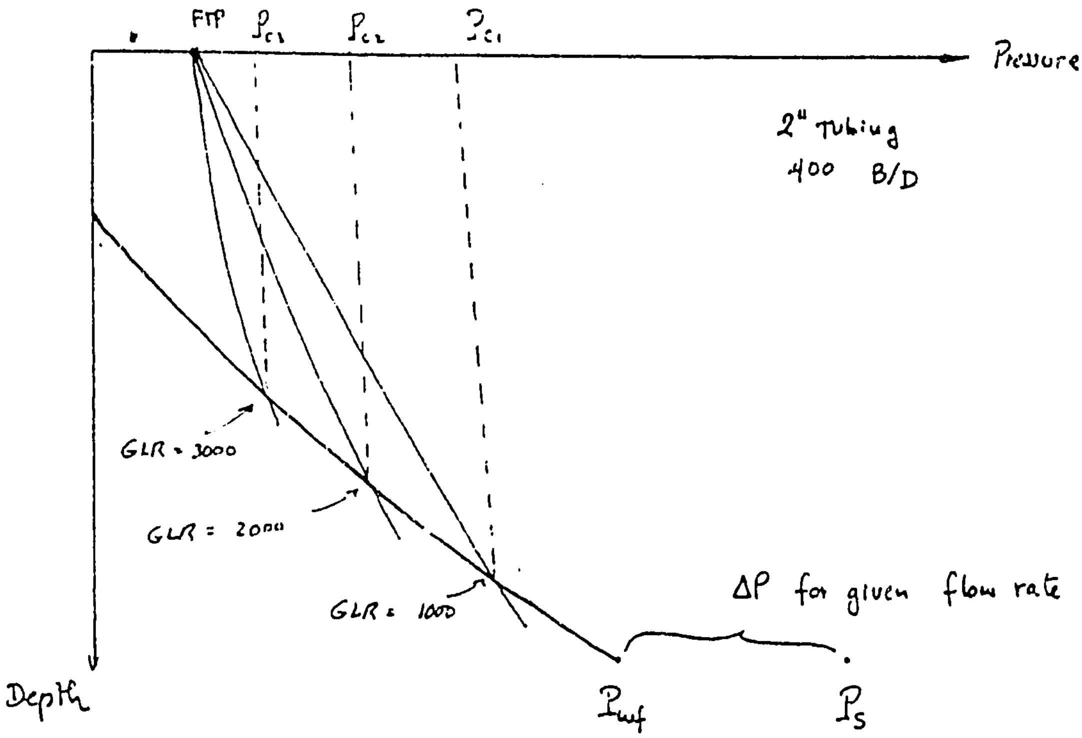
<u>Controllable Tubinghead Pressure</u>

The problem is approached by assuming a tubing size, a desired flowrate and a flowing tubing pressure. The independent variables are therefore casing injection pressure and gas-liquid ratio.

The greater the casing pressure the deeper the point of injection and smaller volume of injected gas to achieve the same flowing tubing pressure. The following figure illustrates this relationship for a given condition.

u(?)

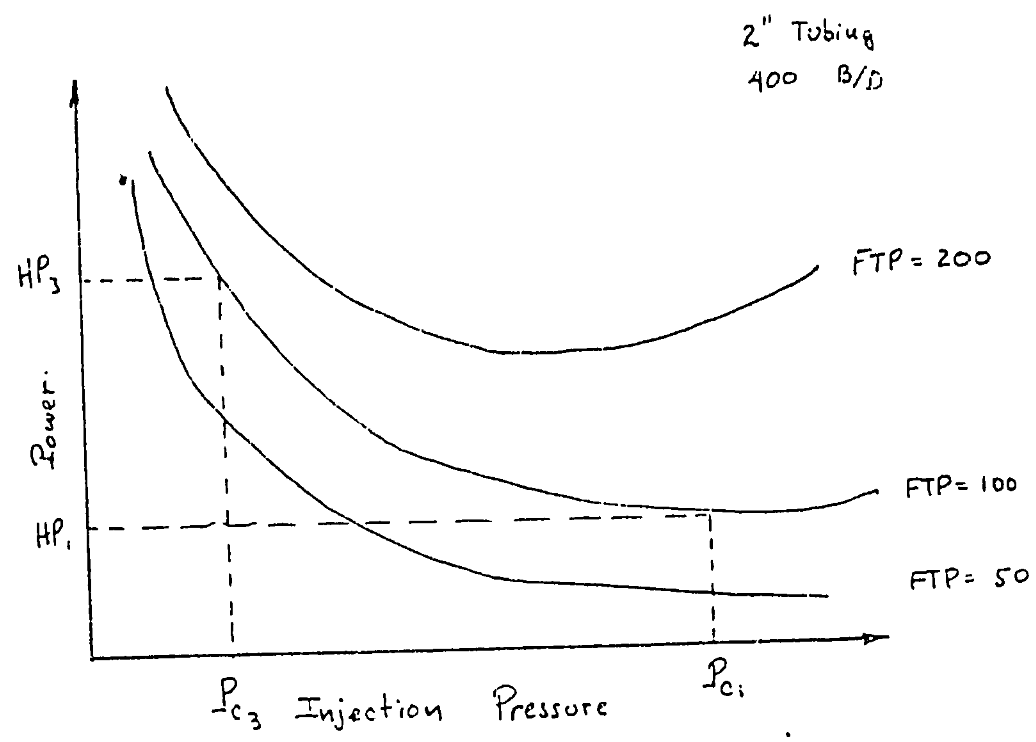
For a given flowing tubing pressure:



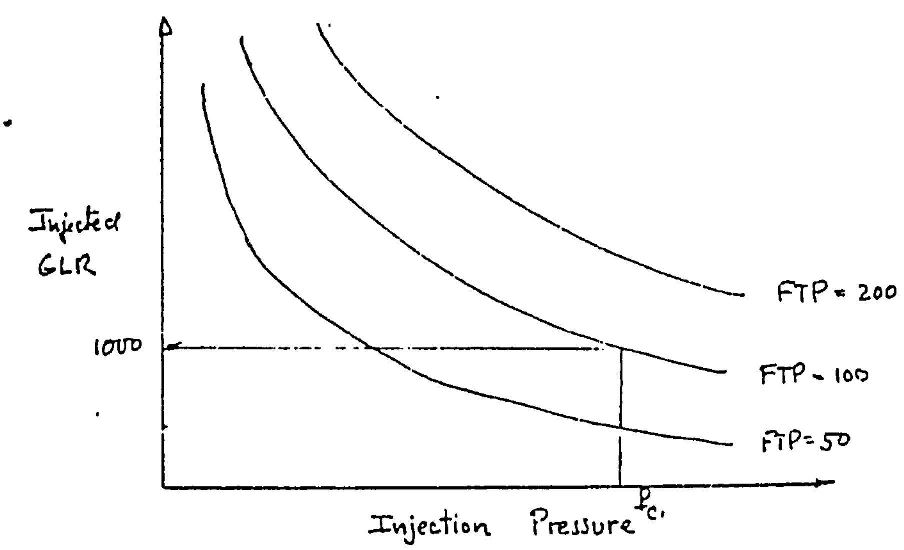
For each case the variable of interest is the power for injection and volume of gas required. Compression horsepower is calculated for each point and the calculation is repeated for various flowing tubing pressures.

This results in families of operating curves which can be used in selecting possible ranges of variables to be considered in detailed economic analysis.

For each point the Adiabatic horsepower is calculated and plotted vs. injection pressure.



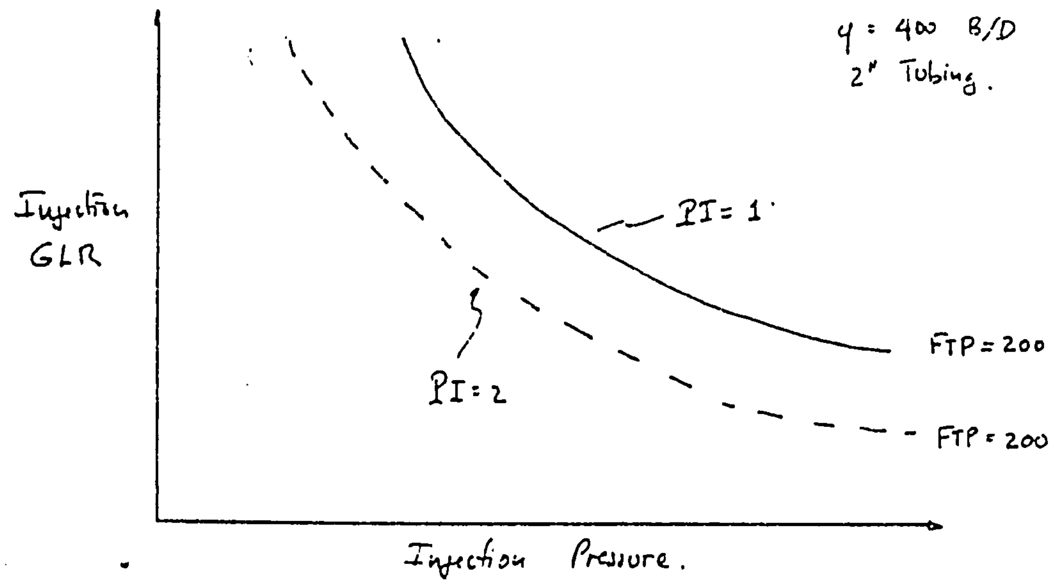
At the corresponding points the injected GLR is plotted vs. njection pressure.



Injection pressure is selected that will result in minimum horsepower over the range of possible flowing tubing pressures, and the corresponding GRL are determined from above.

Note that when various wells are involved having different productivities this will result in different requirements for each well if the same rate is desired for each well.

In this case wells can be grouped in ranges of PI and requirements calculated for these ranges.



Which shows that the gas requirements increase as the productivity decreases, for a given injection pressure.

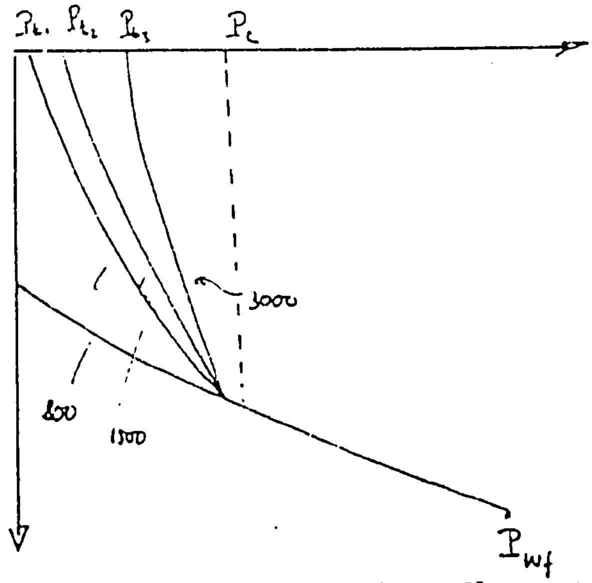
Uncontrollable Flowing Tubing Pressure

The presence of long surface flowlines connecting the wellhead to the separator (constant pressure point) causes a back pressure which is a function of the flowrate through the flowline. Changes in gas liquid ratio will cause changes in tubing pressure. The performance of the system will be controlled by the performance of the flowline and of the wellbore.

Given: Tubing Size

Casing Injection Pressure

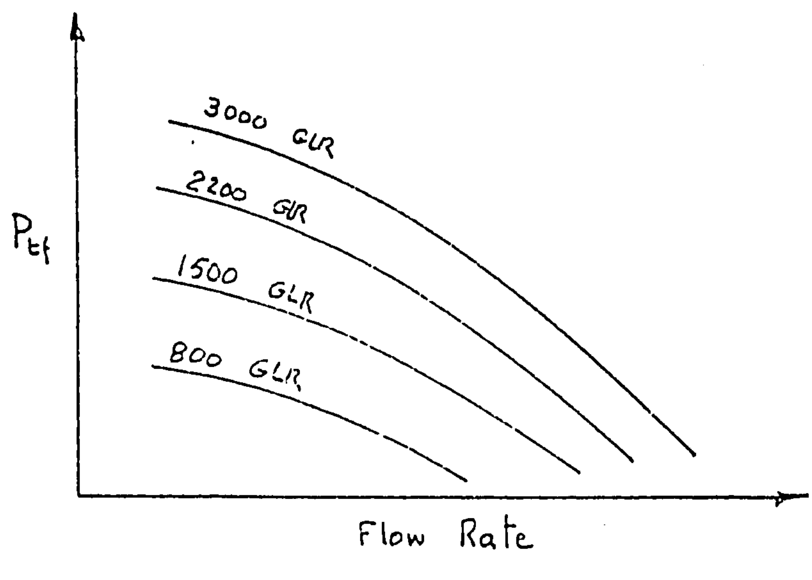
for a flow rate it is possible to establish the various flowing tubing pressures as a function of GLR.

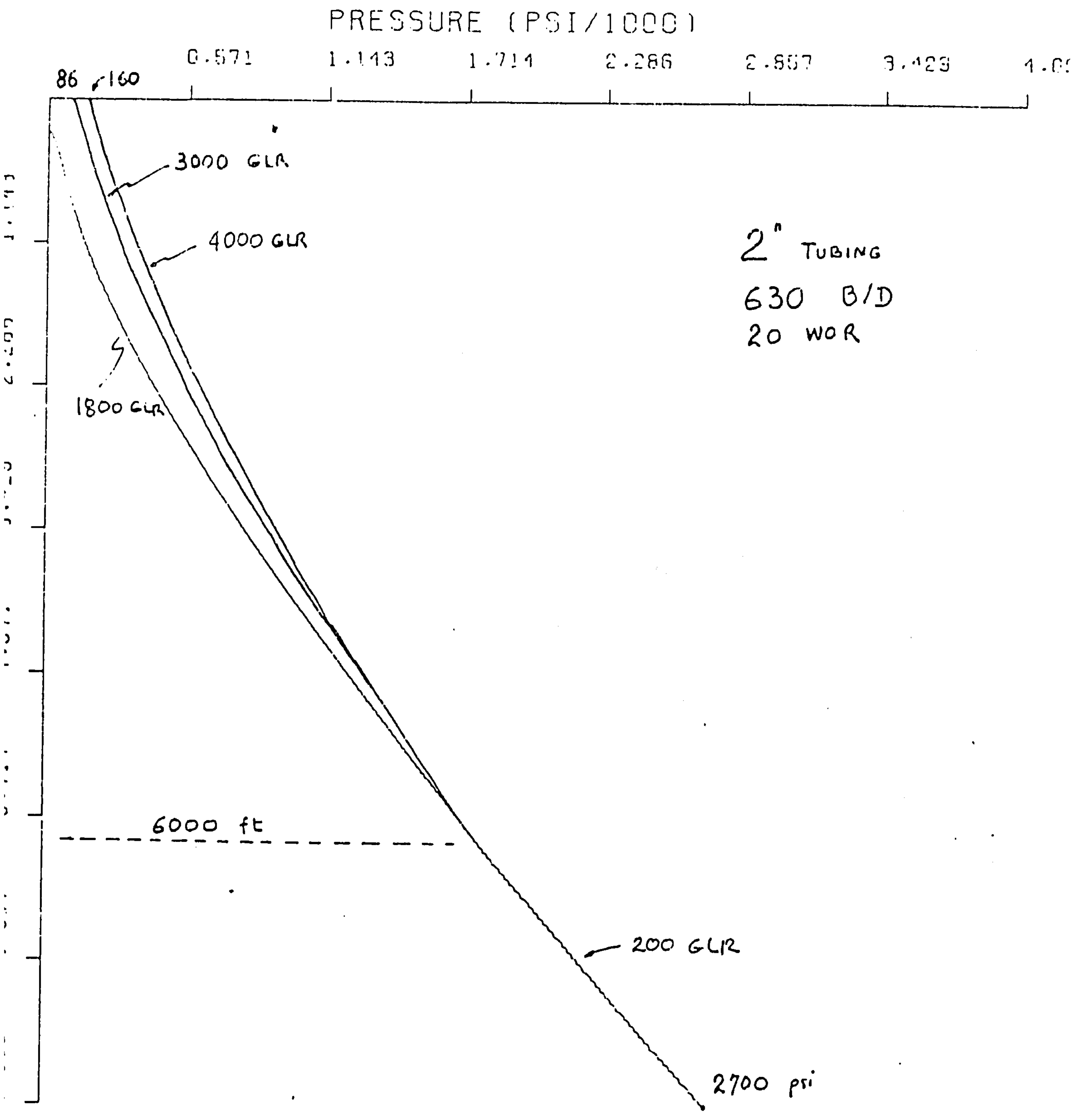


Q = 600 B/D

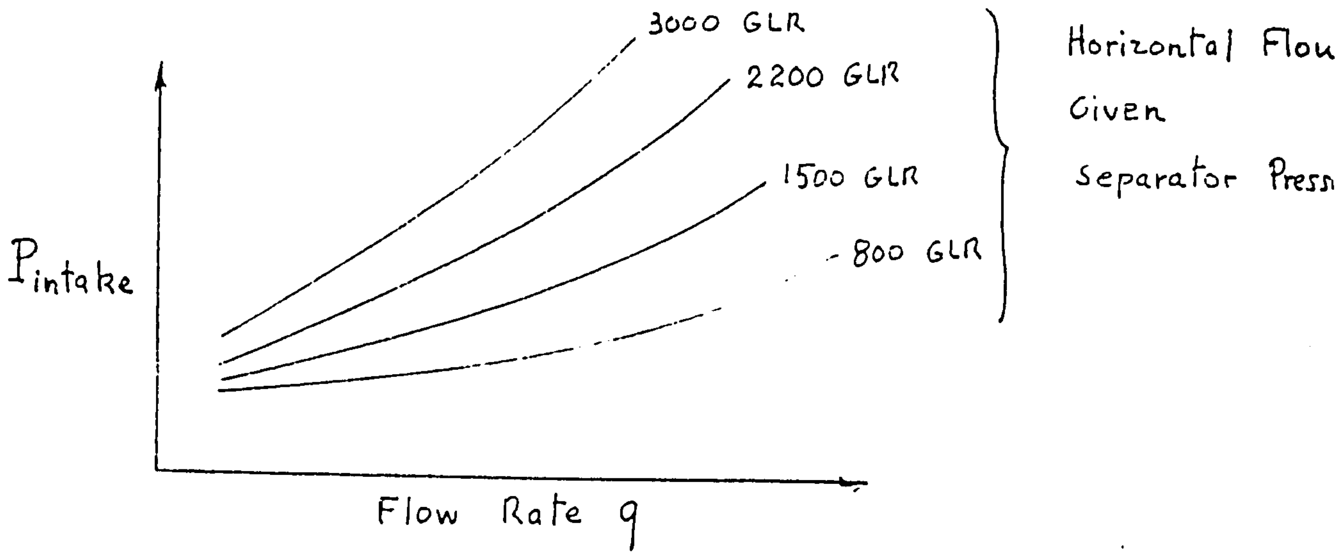
GLR	P_{tf}
800	200
1500	270
2200	235
3000	305
3500	305

repeating this for various flow rates the performance of the wellbore can be obtained as:

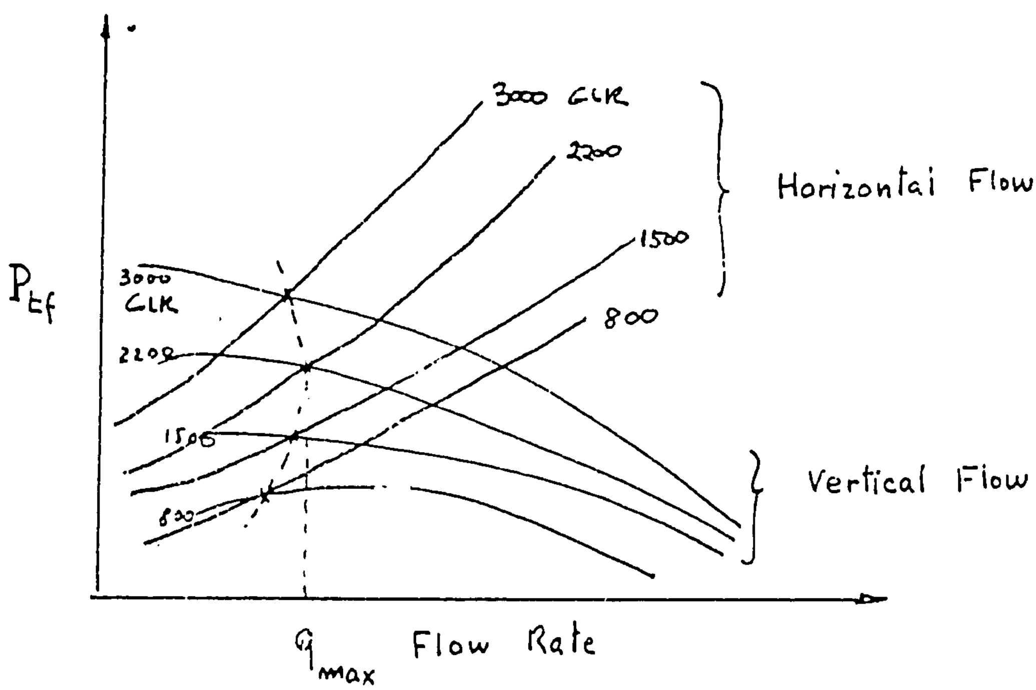



PRESSURE (PSI/1000)
0.571
1.143
1.714
2.286
2.857
3.429
86
160
3000 GLR
4000 GLR
1800 GLR
2" TUBING
630 B/D
20 WOR
6000 ft
200 GLR
2700 psi

For a given flowline site, length and separator pressure, the performance of the flowline can be expressed as a plot of intake pressure vs. flow rate for various gas-liquid ratios:



At steady state conditions $P_{intake} = P_{tf}$ so that the intersection of curves of equal GLR constitutes the possible flow rates as a function of injected gas.



49

The procedure is repeated for available combinations of casing jection pressure, tubing and flowline sizes, resulting in families curves from where operation parameters can be selected for detailed st and efficiency calculations.

dividual Gas Lift Well Design

The majority of these problems involve selecting the depth of e operating valve, the volume of gas to be injected and the spacing the valves used for unloading the well (unloading valves).

ctors That Influence the Design

From the previous discussion can be concluded that the principal ctors affecting the design are:

Available injection pressure

Available gas volume

Tubing size

Flowing tubing pressure

iese parameters must be established prior to undertaking the design.

Generally the following data are needed or appropriate estimates ıve to be made for the unknown quantities.

Depth of well

Depth of tubing

Size of tubing and casing

Size and length of surface flowline

Separator back pressure

Expected flowing tubing pressure

Desired producing rate

Production GOR

Production WOR

Oil, gas, water gravities

Bottom hole temperature

Tubinghead temperature

Well inflow performance or PI

Stabilized formation pressure

PVT data for produced fluids

Injection gas pressure

Injection gas maximum rate

Injection gas PVT data

Kill fluid gradient

Unloading back pressure

Some of the data is seldom known. Generalized correlations can then be used to obtain approximate results.

Design Steps

The following is intended to illustrate one of the many methods. The first part of the design involves the determination of the point of injection.

1. Prepare a pressure vs. depth graph with scales identical to available flowing gradient curves.

2. Plot static BHP.

3. From IPR calculate P_{wf} at the desired flow rate. Plot P_{wf}.

4. Calculate static gradient and plot static gradient line from P_s.

5. Plot flowing gradient line from P_{wf}. Use gradient curves for appropriate rate, GOR, tubing size, etc.

6. Plot casing injection pressure and kick off pressure at surface.

7. Plot gas pressure gradient line in

8. Determine point of balance. (Flowing tubing pressure = casing pressure.)

9. Determine point where $P_{tubing} - P_{casing} = \Delta P_{value}$.

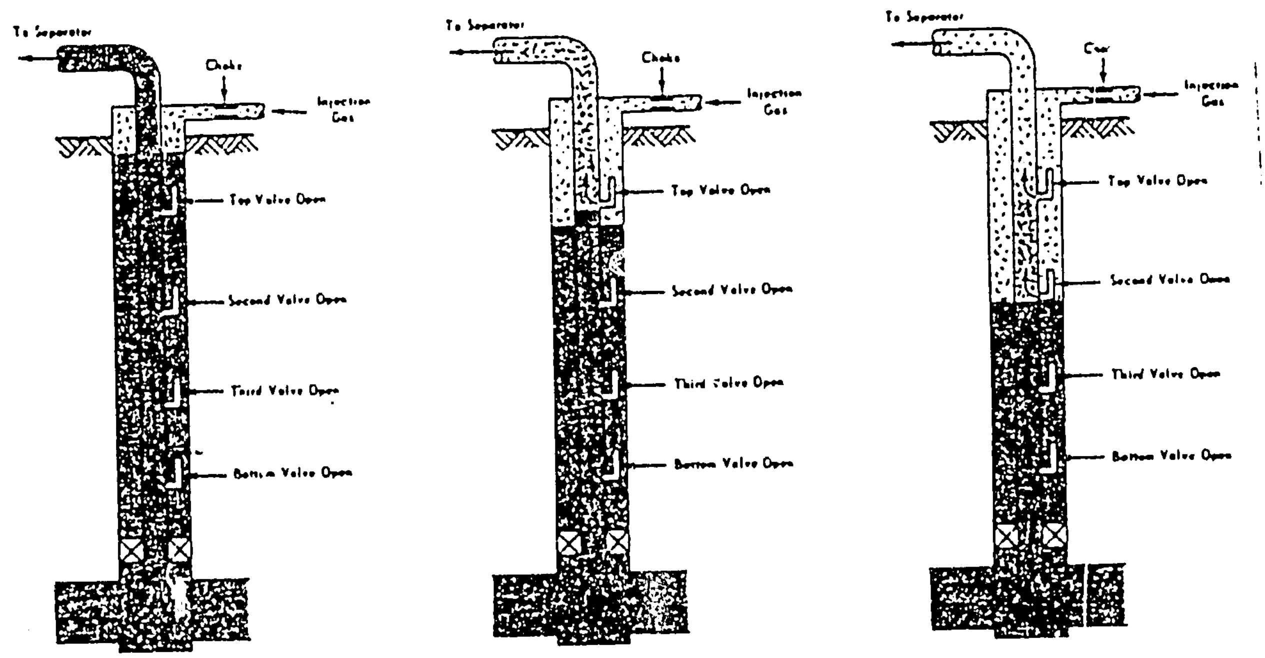
10. Plot flowing tubing pressure (P_{tf}).

11. Connect P_{tf} with point of injection with appropriate gradient line.

These steps yield the depth of the operating valve and the gas-liquid ratio required above the point of injection in order to obtain the desired tubing pressure. The operating valve has to be sized to allow the injection of the gas volume necessary to achieve the design gas-liquid ratio.

The second part of the design involves the determination of the number and spacing of the valves required to unload the well (unloading valves).

The following diagrams illustrate a typical unloading sequence.



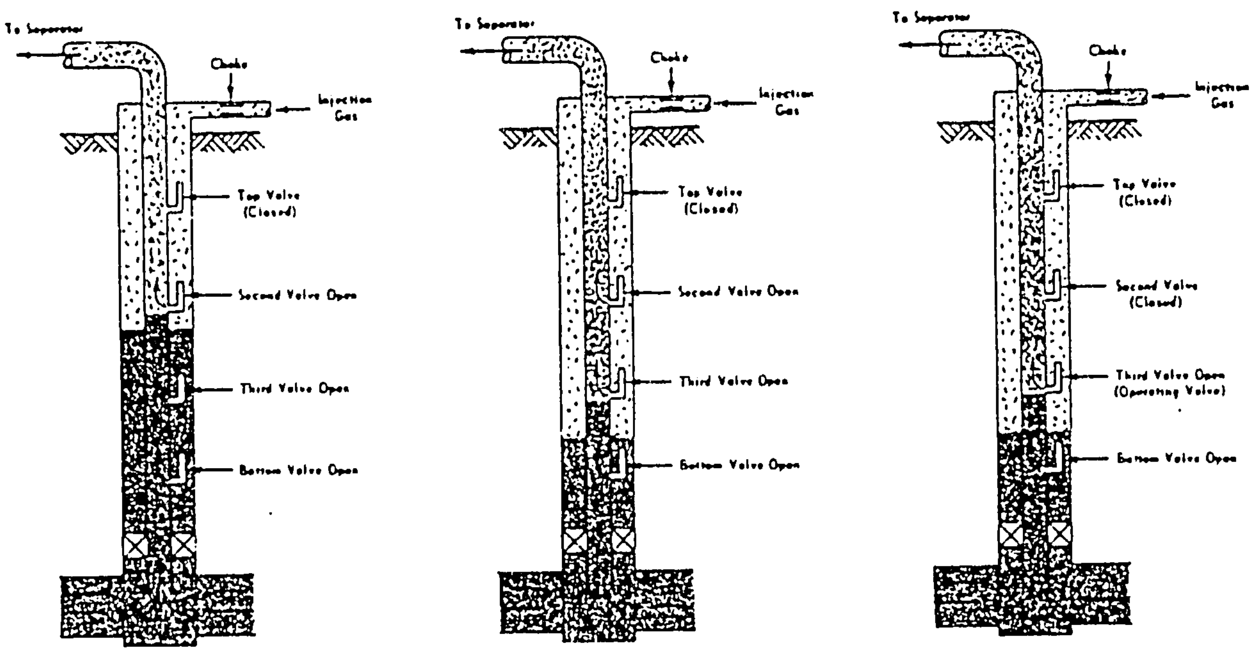
(A) Fluid From Casing Being Transferred Into Tubing Through All Valves And U-Tubed By Injection Gas Pressure To Surface

(B) Fluid In Tubing Being Aerated To Surface By Injection Gas Through Top Valve As Fluid In Annulus Is Transferred Into Tubing Through Lower Valves.

(C) Injection Gas Entering Tubing Through Top And Second Valve Immediately After Second Valve Uncovered

The process aims at reducing the fluid level in the annulus until the operating valve is uncovered.

The important aspect is that at any time only one valve should be open and injecting gas. If this is not the case the efficiency of the installation is greatly reduced and it may not be possible to uncover the operating valve. Flowing bottom hole pressure will be greater than that required to produce the desired liquid.



(D) Fluid In Tubing Being Aerated To Surface By Injection Gas Through Second Valve As Fluid In Annulus Is Transferred Into Tubing Through Third And Bottom Valves

(E) Injection Gas Entering Tubing Through Second And Third Valves Immediately After Third Valve Is Uncovered

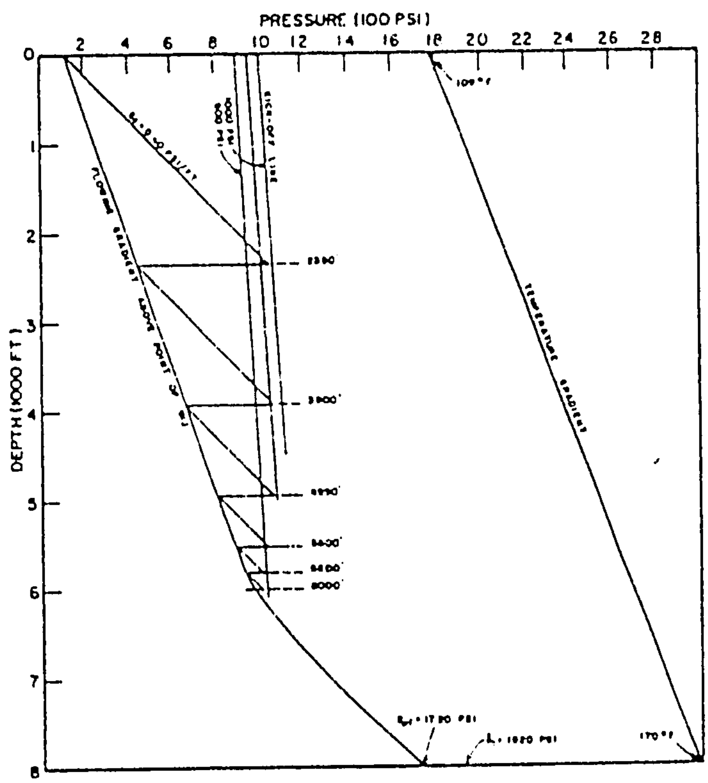
(F) Producing Rate Equals Capacity Of Tubing From Third Valve For Available Injection Pressure Therefore, Bottom Valve Cannot Be Uncovered

Spacing of valves depends greatly on the type of valve used (pressure valve, fluid valve, balanced valve, etc.) and the parameters reflecting gas availability, maximum injection pressure and tubing back pressure.

The following diagram illustrates one such procedure which assures:

Unlimited kick off gas available

Pressure valves



Valve Mechanics

The following is a brief outline of the principal characteristics of the major types of gas lift values:

Continuous Flow: Capable of throttling gas into the tubing string keeping pressure constant inside tubing. Change orifice size to take care of injection rate changes.

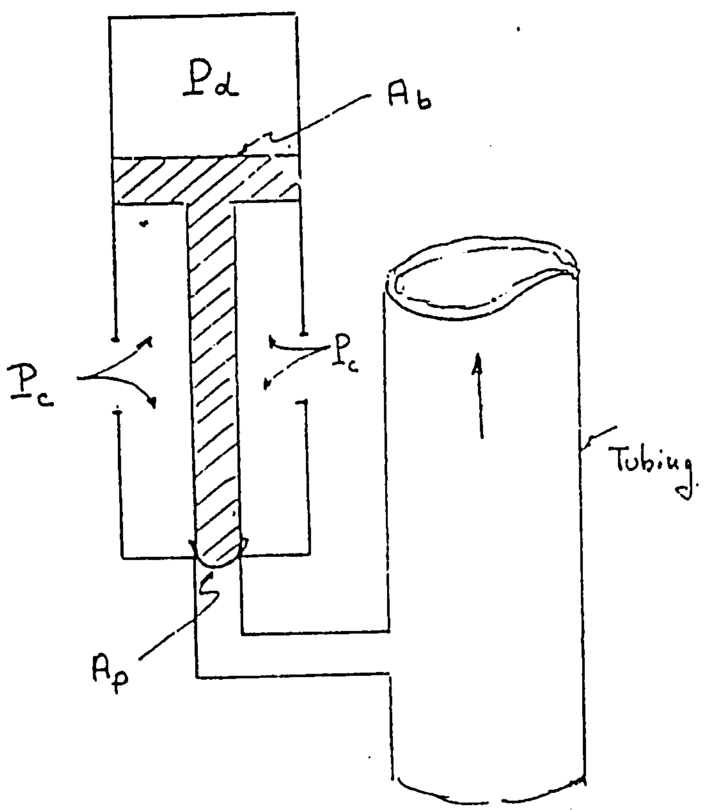
Intermittent Flow: Large port size to allow quick injection of gas into tubing. ¼ - 1" port sizes.

Either can be opened by:

1. buildup of pressure in annulus
2. buildup of pressure in tubing
3. combination of 1 and 2

Bellows Type Valves

A. Unbalanced



Intermittent flow

Forces closing valve

$$F_c = P_d A_b$$

Forces opening valve

$$F_{open} = P_c (A_b - A_p) + P_t A_p$$

<u>Valve closed</u> ready to open

$$F_c = F_o$$

$$P_d\ A_b = P_c\ (A_b - A_p) + P_t\ A_p$$

$$P_{c/open} = \frac{P_d - P_t\ (A_p/A_b)}{1 - A_p/A_b}$$

$$\text{Let } \frac{A_p}{A_b} = R$$

$$P_{c/open} = \frac{P_d - P_t\ R}{1 - R}$$

R is also known as the tubing effect.

<u>Valve open</u> ready to close

Force to close = $F_c = P_d\ A_b$

Force to open = $F_o = P_c\ (A_b - A_p) + P_c\ A_p = P_c\ A_b$

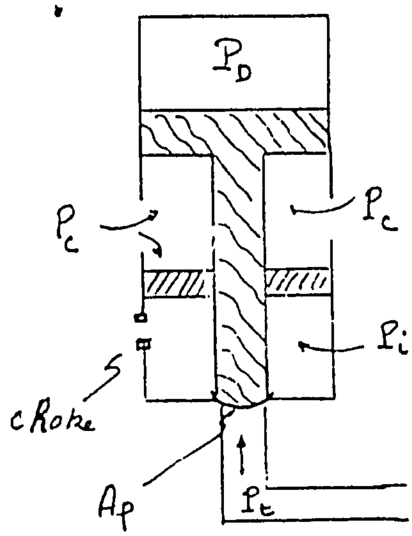
Closing valve $P_c = P_d$

Assume $R = 0.1$

$P_d = 700$ psig

$P_{c/open}$	$P_{c/close}$	P_t	Spread
777	700	0	77
755	700	200	55

Continuous Flow Valve



Closing

$P_d\ A_b$

Opening

$P_c\ (A_b - A_p) + P_t\ A_p$

$$P_{c/open} = \frac{P_d - P_t\ R}{1 - R}$$

After opening, the pressure below the stem will be different (less) than P_c.

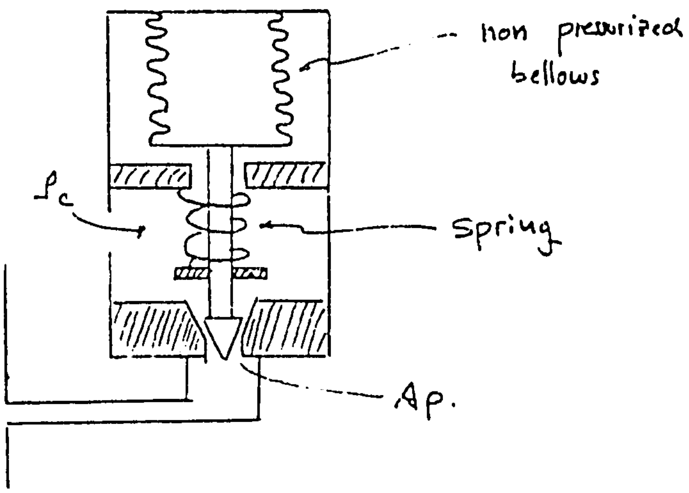
Open to close

$$P_c\ (A_B - A_p) + P_i\ A_p = P_d\ A_b$$

$$P_c = \frac{P_d\ A_b - P_i\ A_p}{A_b - A_p} = \frac{P_d - P_i\ R}{1 - R}$$

P_i will be a function of tubing pressure.

Variable Choke Valve

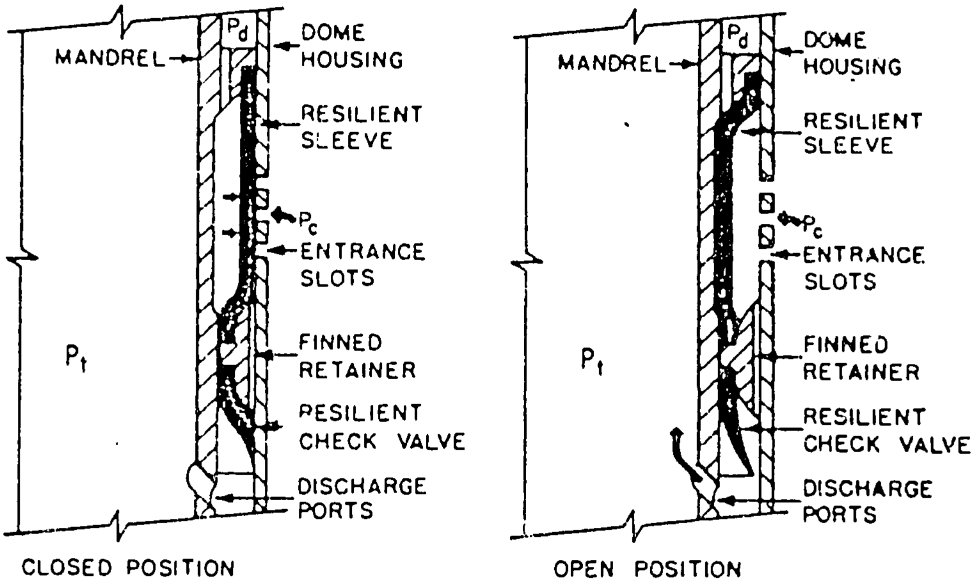


B. Balanced

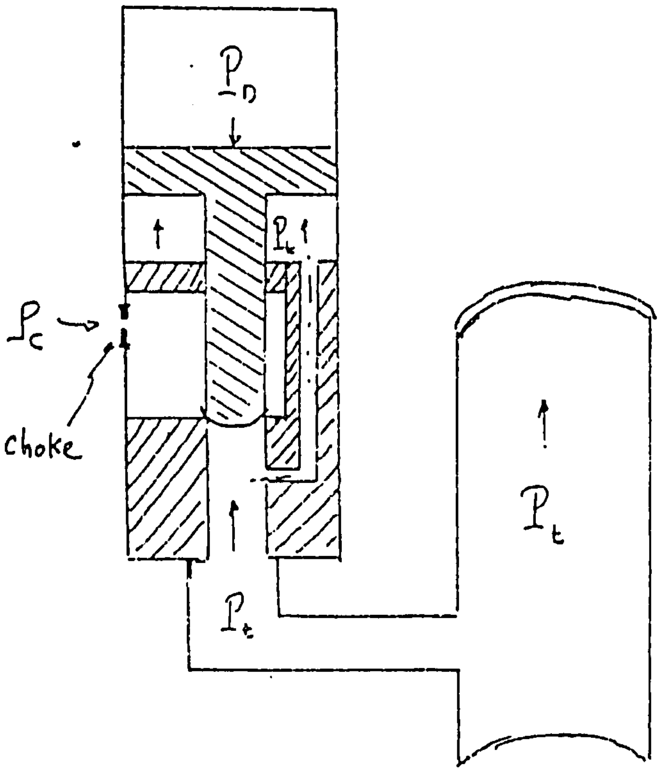
Flexible Sleeve

$P_{c/o} \geq P_d$

$P_{c/c} \leq P_d$



C. Fluid Operated Valves (Balanced)

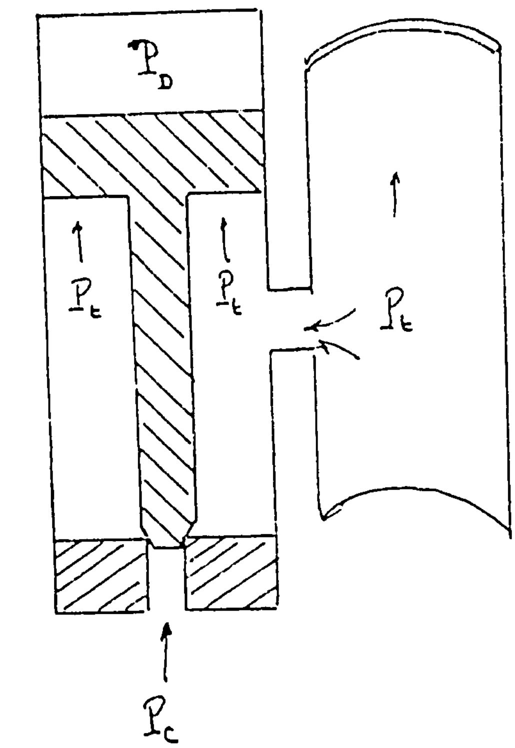


Closing = $P_D\ A_B$

Opening = $P_t\ A_p + P_t\ (A_B - A_p)$

$P_D = P_t$

Fluid Operated Valve (Unbalanced)



Closed to open

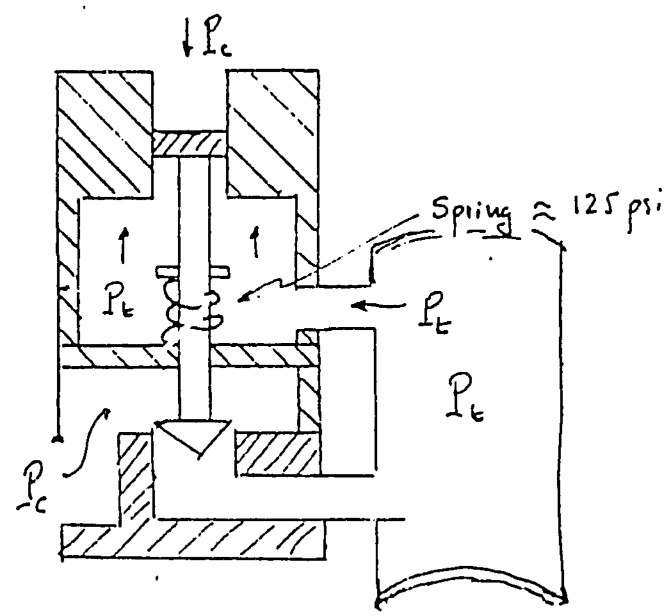
Closing $P_D\ A_B$

$$\text{Opening} = P_t\ (A_B - A_p) + P_c\ (A_p)$$

$$P_d = P_t\ (1 - R) - P_c\ R$$

Also fluid operated valves with uncharged bellows and spring load.

Differential Valves



$$P_c = P_t + 125\ \text{psig}$$

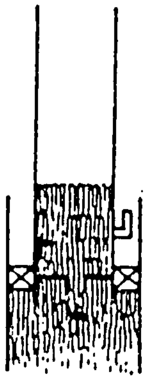
The spring controls the difference in pressure between casing and tubing at which the valve opens.

Example: Data

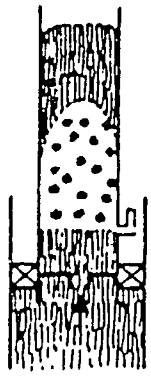
Depth of Well	8000 ft.
Size of Tubing and Casing	2" tubing
Producing Conditions: Sand, Paraffin	
Size and Length of Surface Flowline	
Separator Back Pressure	50 psig
Expected Flowing Tubing Pressure	100 psig
Desired Producing Rate (Total Fluid)	600 Bbl/day
% Water	95%
S.G. of Injection Gas	0.65
Injection Gas Pressure and Volume	900, no limit
IPR	PI = 3
BHT	210°F
Surface Flowing Temperature	150°F
API Gravity of Oil	40°API
S.G. of Water	1.02
Solution GOR	200 SCF/Bbl
Static BHP	2900 psig
F.V.F.	
Viscosity and Surface Tension	
Kill Fluid Gradient	0.5 psi/ft
Loaded to Top	
Unloading to pits - first valve to sep. - all others P_t = 100 psig	
P_{Ko}	100 spig
Pressure Valves	25 psi casing pressure drop per valve

Intermittent Lift

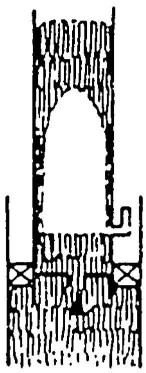
Fluid produced from the formation is allowed to accumulate in the tubing. Gas is then injected at a high rate into the tubing with the objective of displacing the liquid slug to the surface as shown in the following diagram.



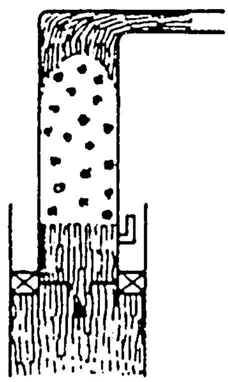
A. Buildup of Liquid Slug



B. Fallback as Liquid Droplets Below Slug



C. Fallback on Tubing Wall Below Slug



D. Valve Closed

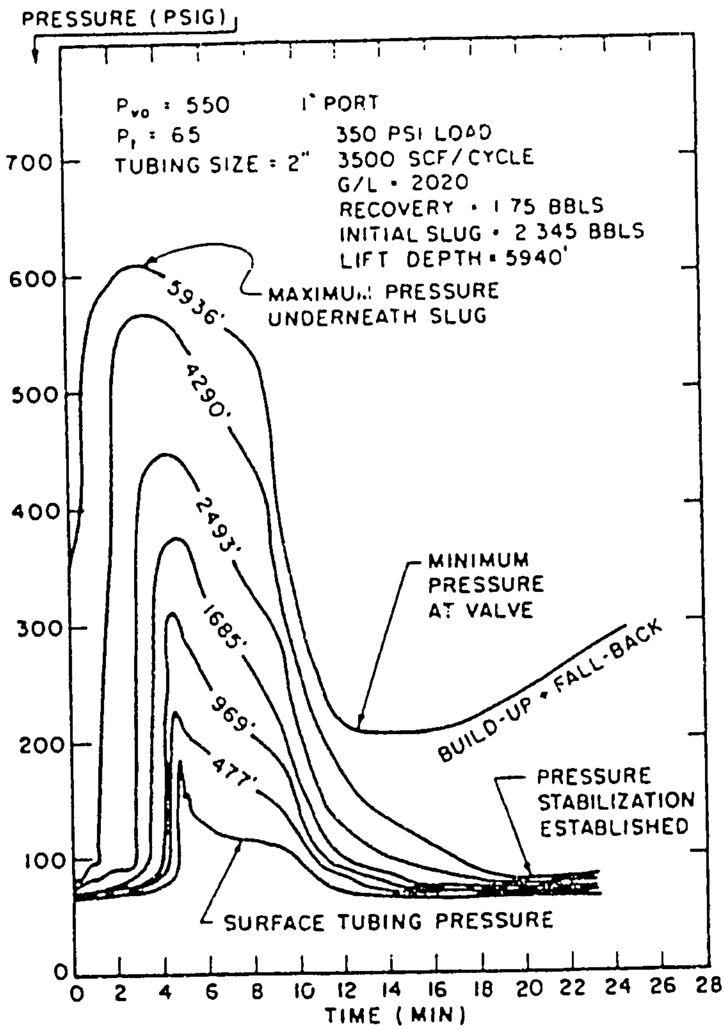
When the combination of surface back pressure, weight of gas column and hydrostatic pressure of the slug reaches a specified value at the gas lift valve, gas is injected into the casing annulus through some type of control at the surface for a definite injection time.

When the casing pressure increases to the opening pressure of the gas lift valve, gas is injected into the tubing string. Under ideal conditions the liquid, in the form of a slug or piston, is propelled upwards by the energy of the expanding and flowing gas beneath it. The gas travels at an apparent velocity greater than the liquid slug velocity, resulting in penetration of the slug by the gas. This penetration causes part of the liquid slug to fall back into the gas phase in the form of droplets and/or as a film on the tubing wall.

When the liquid slug is produced at the surface, the tubing pressure at the valve decreases, increasing the gas injection through the valve. When the casing pressure drops to the valve closing pressure, gas injection ceases. Following production of the slug, a stabilization time occurs during which the fallback from the previous slug falls or flows to the bottom of the well and becomes a part of the next slug, which is feeding in from the producing zone.

Liquid fallback can represent a substantial part of the original slug. Control of fallback determines the success of an intermittent gas lift installation. The inability to predict liquid fallback has resulted in overdesign of many installations. In many cases high recovery rates are achieved, but frequently at excessive operational costs which limit the profit-making ability of the wells.

The following figure shows a typical recording in which gas was injected into 2" tubing at a depth of 5940 feet through a 1" ported gas lift valve. Pressure recordings are illustrated at depths of 5936, 4290, 2493, 1685, 967, 477 and 0 feet. The initial tubing pressure at the valve was 350 psi, and the initial slug volume was 2.345 B(95% salt water).



The following information can be obtained from the pressure recording at the valve at 5936 feet:

1. At zero time the initial tubing load was 350 psi.

2. As gas was injected, the slug accelerated until the tubing pressure at the valve reached 600 psi within 2-3 minutes.

3. The slug reached the surface in 4 minutes, 35 seconds as noted on the zero depth curve (surface tubing pressure). The pressure at 5936 feet began to decrease at this time, although the gas lift valve had not yet closed.

4. As the slug was produced at the surface, the pressure at 5936 feet dropped to approximately 530 psi in 8 minutes, at which time the gas lift valve closed.

5. The pressure then dropped sharply to a minimum of 208 psi in about 12 minutes. The minimum pressure represents a combination of well back pressure, liquid fallback, and fluid feed-in into the wellbore from the producing formation.

6. The minimum pressure remained fairly constant for 4-5 minutes during which time the fluid in droplet form was still being produced at the surface, tending to reduce the pressure at 5936 feet. However, liquid fallback and feed-in offset the pressure reduction, resulting in the constant pressure. The shape of the curve depicting minimum pressure before the pressure builds up varies, depending upon the rate of liquid production (82 B/D for this well).

7. At approximately 18 minutes, the pressure at 5936 feet began to show a decided increase due to liquid feed-in.

From the pressure recording at 4290 feet, observations similar to those at 5936 feet can be made:

1. The stabilized tubing pressure at zero time was 80 psi, indicating that the top of the slug was initially below this point.

2. The top of the slug reached 4290 feet in about one minute, at which time the pressure began to increase.

3. The pressure continued to increase as the slug passed 4290 feet, but did not reach the pressure level of 600 psi attained at 5936 feet. The lower maximum pressure of 575 psi is a result of part of the slug being lost as liquid fallback.

4. The pressure dropped at about the same rate as on the 5936 feet recording when the slug was being produced at the surface and after the gas lift valve closed.

5. After 12 minutes the pressure at 4290 feet continued to fall since liquid feed-in had not reached this level.

6. After approximately 18 minutes, the pressure had again stabilized at 80 psi. This represents the time required for the fallback in the tubing to settle completely,

and is important in determining the optimum physical cycle frequency.

The pressure recordings at 2493, 1685, 969 and 477 feet show the same general trends as do those at 4290 and 5936 feet. The times at which the slug reached these depths can be easily determined. After attaining their maximums, the pressures continually decreased, becoming constant after about 18 minutes. With decreasing depth, lower pressure maximums are discernible, indicating more liquid fallback.

The pressure curve at the surface (zero depth) shows the following:

1. At zero time the tubing back pressure was 65 psi.

2. The liquid reached the surface at 4 minutes, 35 seconds.

3. The pressure reached a maximum of 195 psi.

4. The pressure dropped immediately, indicating the major portion of the production had been recovered. Subsequent production in the form of droplets and finely-dispersed fluids follow production of the intact slug.

5. The gas following the liquid slug (tail gas) has completely escaped within 14 minutes.

Pumping

There are three major types of downhole pumping systems:

a) Sucker rod pumping

b) Electrical submersible centrifugal pumping

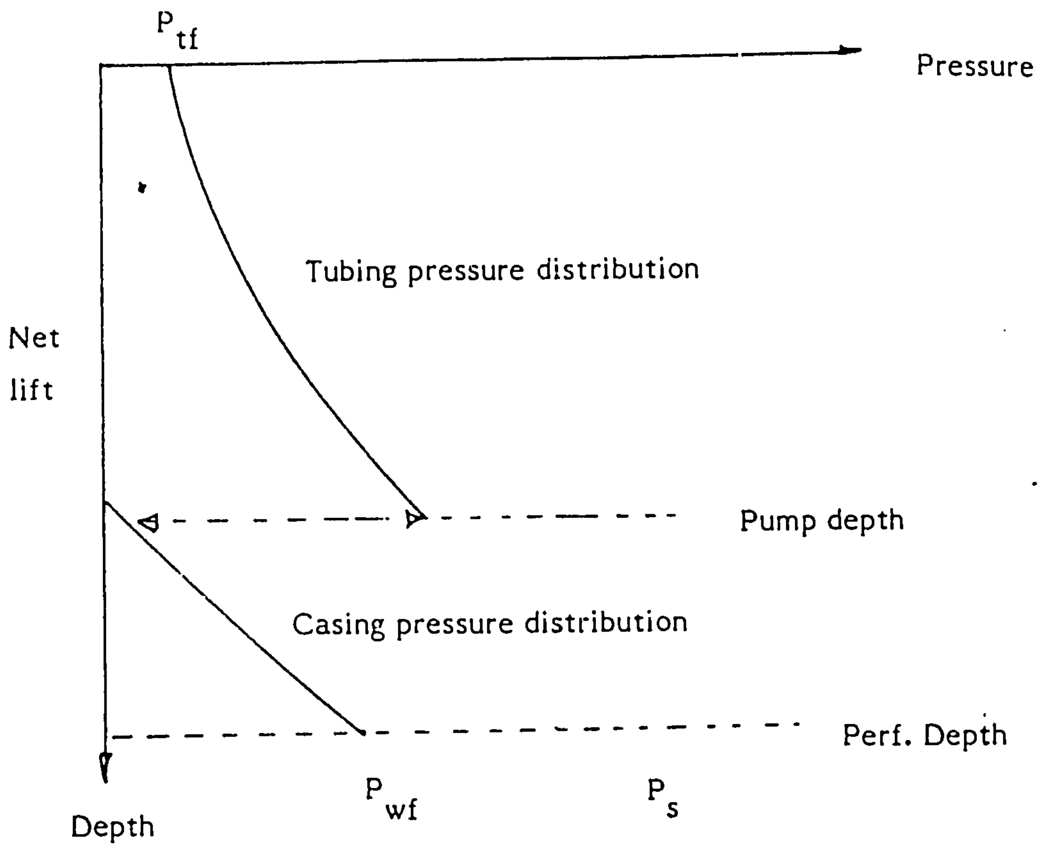
c) Hydraulic

Their relative importance is approximately such that of all the U. S. wells producing by artificial lift ($\approx$ 92% of all U. S. producing wells) about 85% use rod pumping, 2% submersible and 2% hydraulic with the remaining 11% being produced by gas lift. The majority (93%) of the rod pumping wells are strippers (less than 10 B/day) although they usually produce greater volumes of fluid because of large water to oil ratios.

General Concepts

In all cases the pump provides energy to move fluid to the surface allowing the use of reservoir energy only to move fluid to the wellbore and up to the pump intake.

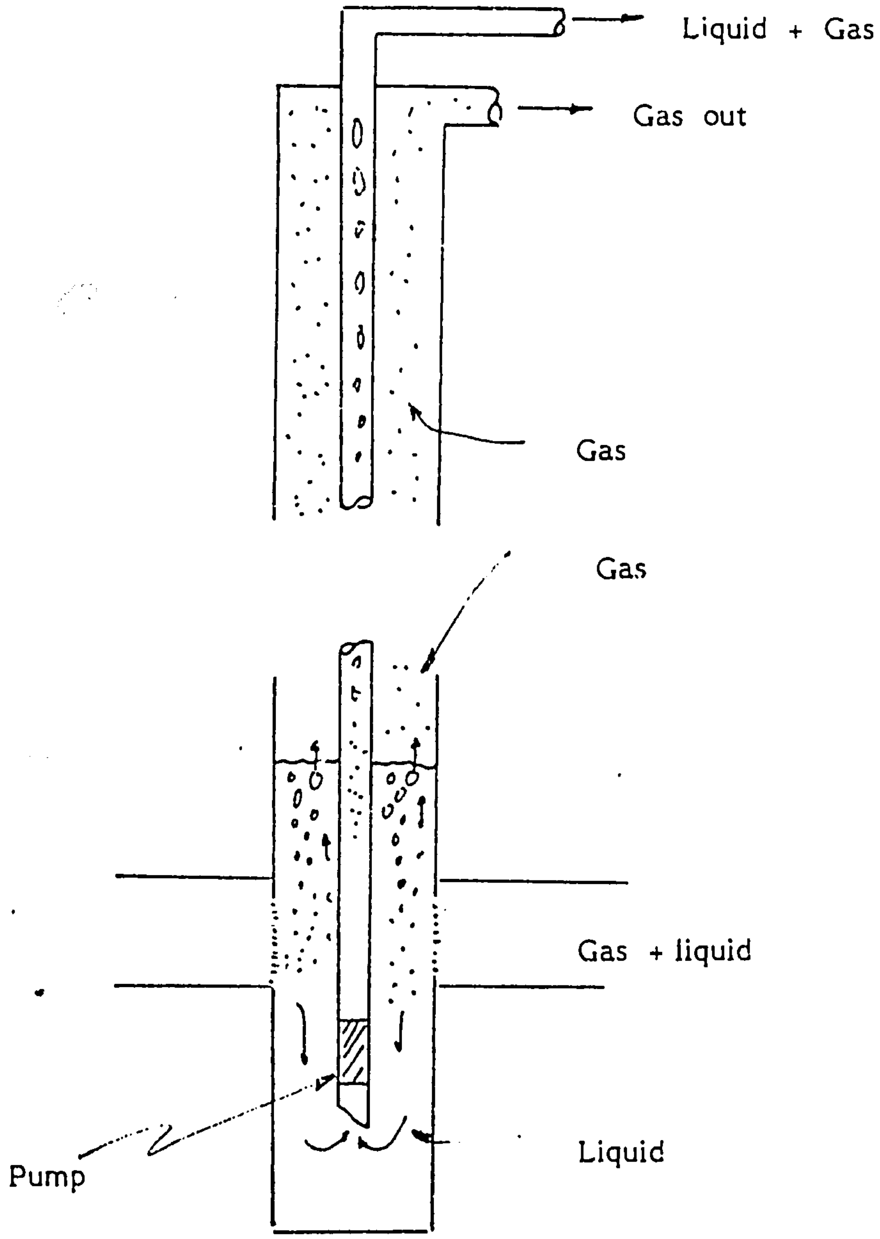
A pressure-depth diagram for a pumping system will be similar to the following.



The pump will be set at a depth greater than the liquid level in the casing annulus to insure that sufficient head is available to flow into the pump intake.

Pump displacement has to be adequate in relation to pump depth and formation productivity. If the pump capacity is too large the fluid level will drop to the pump level. Gas will enter the pump, reducing its efficiency and possibly damaging it. If the pump capacity is too small the fluid level will rise above that required to maintain the appropriate drawdown $(P_r - P_w)$ and it will not be possible to achieve the desired production.

The efficiency of any pumping system is greatly reduced by the presence of gas in the flowing stream. Whenever possible pump depth should be such that the intake pressure is greater than the bubble point pressure of the fluid being pumped. For most applications this can only be achieved by letting the majority of the gas in solution evolve and rise through the annulus to be produced at the casing head.



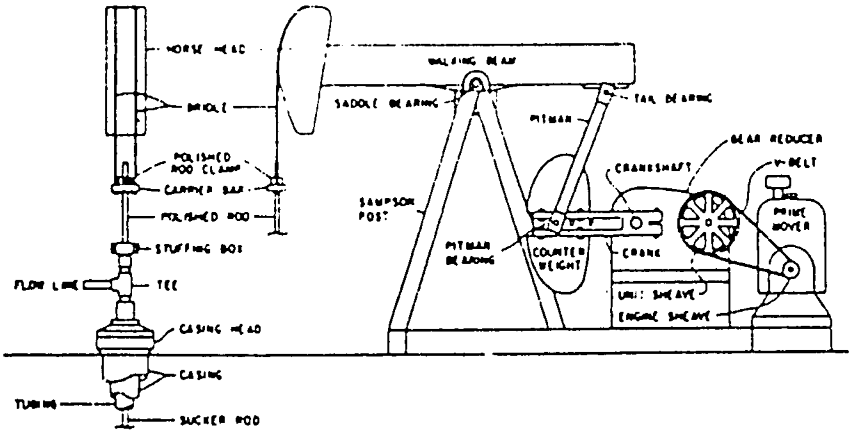
Sucker Rod Pumping (also Beam Pumping)

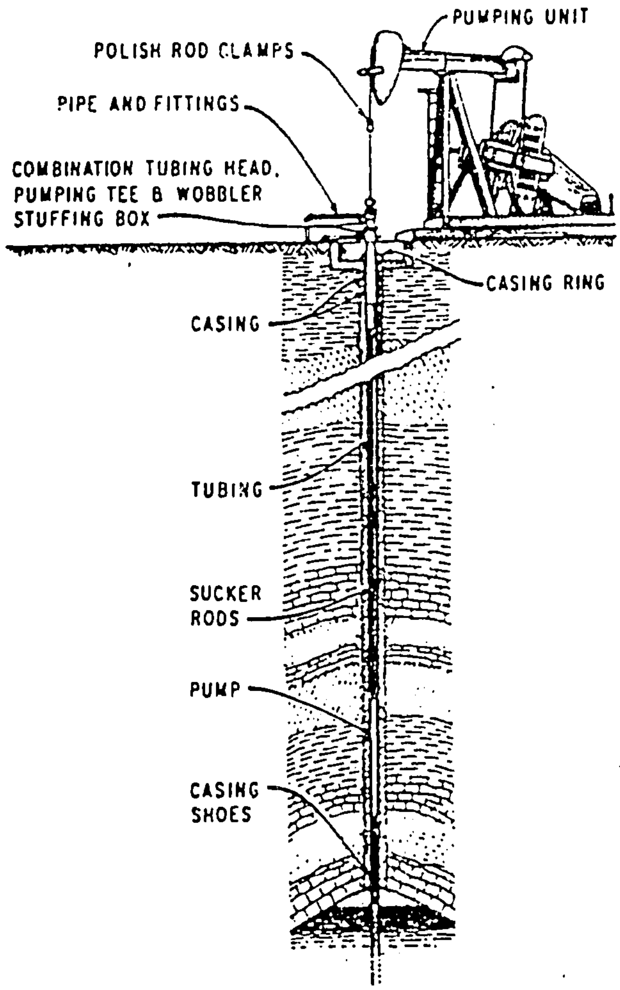
Steel (also fiberglass) rods are used to transmit reciprocating motion to the downhole positive displacement pump from the surface beam pumping unit.

System components:

a) Pumping unit
   - Prime mover
   - Gear reducer
   - Beam
   - Counterbalance
b) Sucker Rod string
c) Downhole pump
d) Tubing anchor, gas anchor, polished rod, stuffing box, etc.

The following schematic diagram indicates the relative position of the principal elements.


HORSE HEAD
BRIDLE
POLISHED ROD CLAMP
CARRIER BAR
POLISHED ROD
STUFFING BOX
FLOW LINE
TEE
CASING HEAD
CASING
TUBING
SUCKER ROD
WALKING BEAM
SADDLE BEARING
TAIL BEARING
PITMAN
GEAR REDUCER
CRANKSHAFT
V-BELT
PRIME MOVER
SAMPSON POST
PITMAN BEARING
COUNTER WEIGHT
CRANK
UNIT SHEAVE
ENGINE SHEAVE


PUMPING UNIT
POLISH ROD CLAMPS
PIPE AND FITTINGS
COMBINATION TUBING HEAD,
PUMPING TEE & WOBBLER
STUFFING BOX
CASING RING
CASING
TUBING
SUCKER
RODS
PUMP
CASING
SHOES

The Pumping Problem

Production engineers are faced with two types of problems with regard to rod pumping:

a) Design problem

b) Performance problem

Design problem consists either of the complete design of a pumping system:

Select pumping unit, rods, pump, speed, stroke

or for a given pumping unit:

Select rods, pump, speed, stroke.

In either case the design objective will be to produce at a certain oil flow rate.

The Performance problem involves analysis of an existing pumping system to determine if it is operating according to design and if not recommend necessary changes.

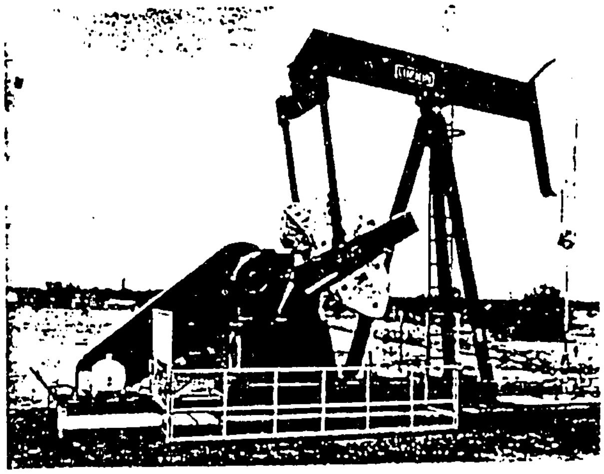
Description of Components

a) Pumping Units

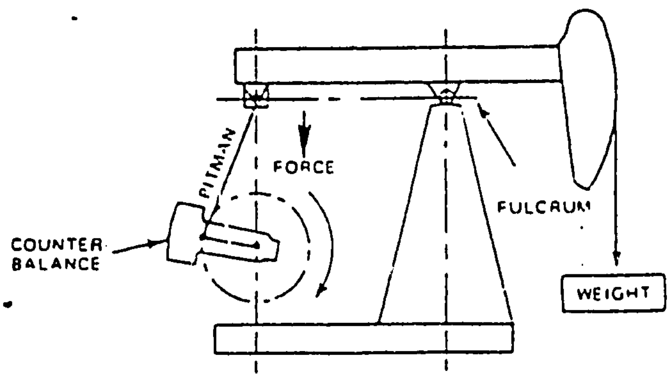
Pumping units are classified according to type:

Unit Type	API Designation
Conventional or Crank Counterbalance	C
Beam Counterbalance	B
Air Counterbalance	A
Mark II (Unitorque)	M
Long Stroke	--

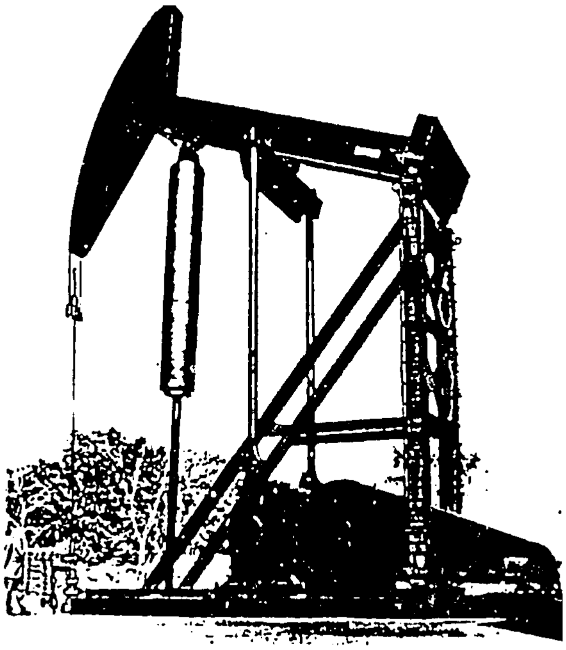
The following figure shows the various types of pumping units.



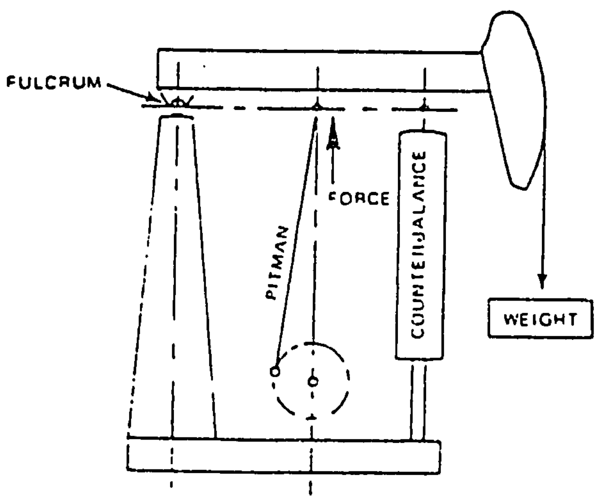
CONVENTIONAL UNITS



CLASS I LEVER SYSTEM - CONVENTIONAL UNIT.

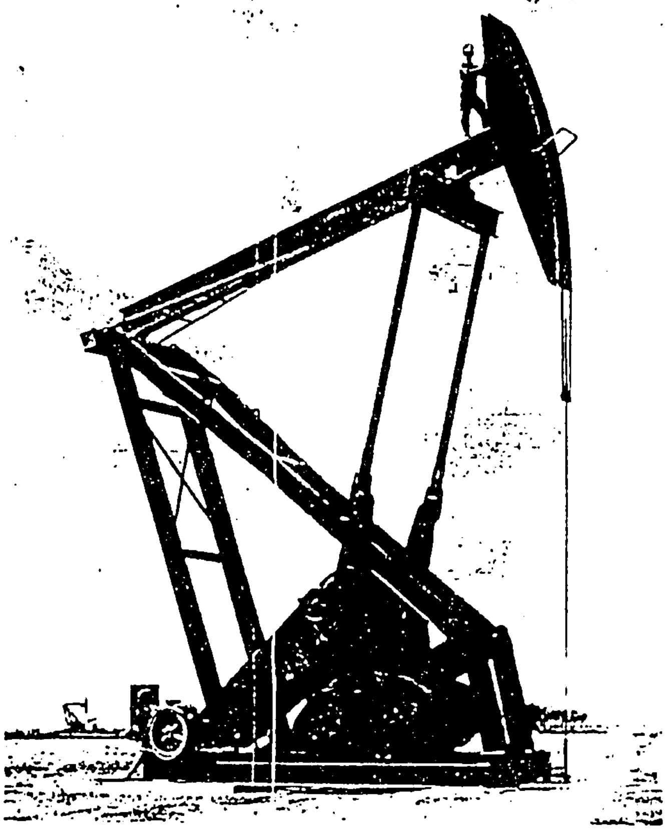


AIR BALANCED UNITS

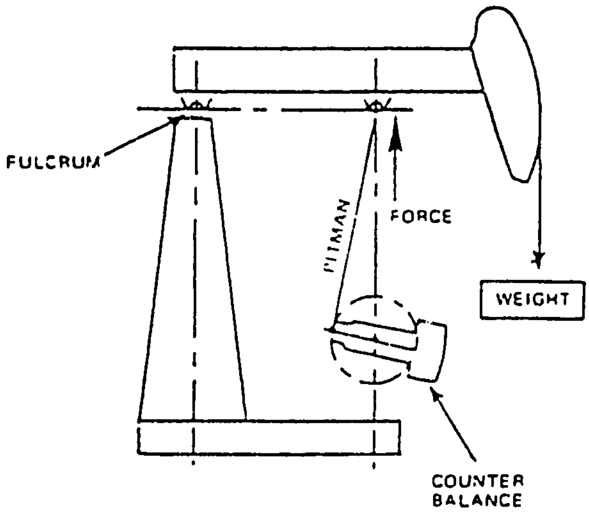


CLASS III LEVER SYSTEM - AIR BALANCED SYSTEM.

76



MARK II UNITORQUE UNITS



CLASS III LEVER SYSTEM - LUFKIN MARK II.

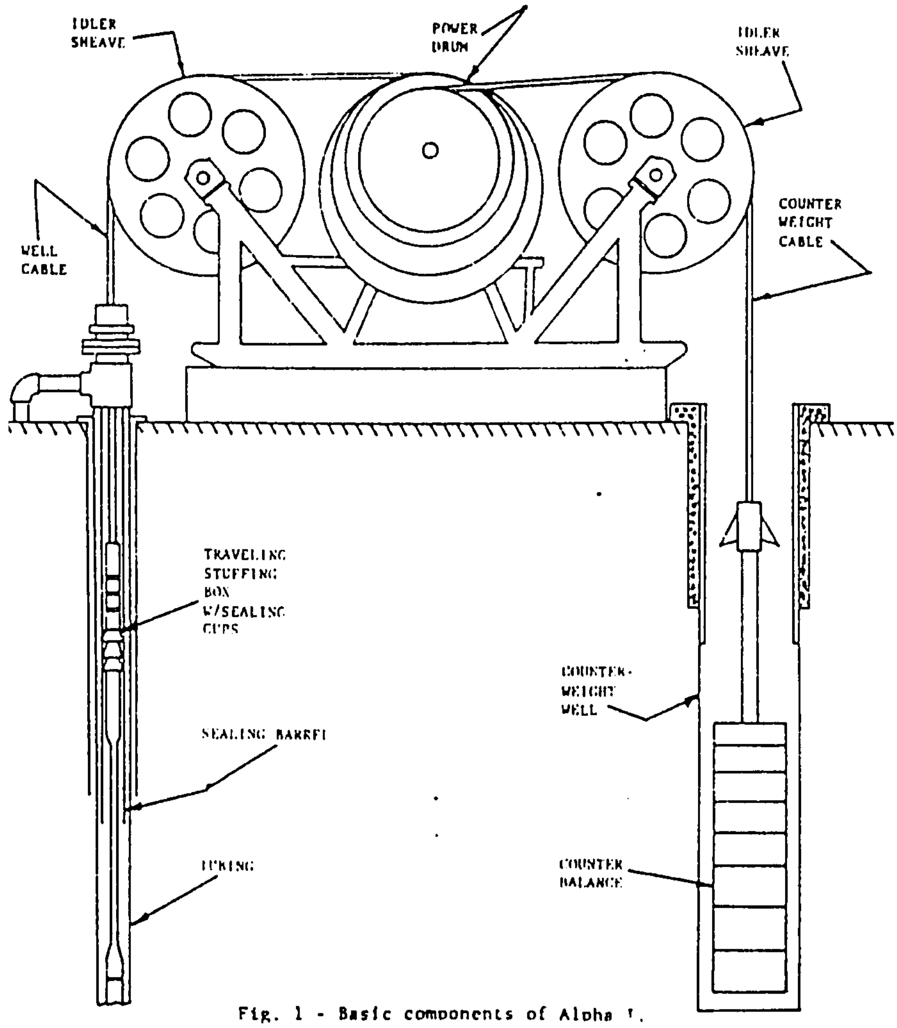


Fig. 1 - Basic components of Alpha I.

Long Stroke Pumping Unit

Pumping Unit Rating is defined by three parameters:

a) Peak torque - that can be developed at the gear reducer (in-lb)

b) Beam Load - that can be applied to the polished rod (lbs)

c) Maximum stroke - that can be transmitted to pump.

The following table presents the typical range of above parameters corresponding to the various pumping unit types currently manufactured.

Unit Type	Torque-Range in.-lb	Beam Load Range lbs	Stroke Range in.
C	5,000 - 912,000	5,300 - 16,800	30 - 168
B	4,000 - 57,000	7,600 - 47,000	64 - 300
A	11,000 - 3,648,000	17,300 - 47,000	64 - 300
M	80,000 - 1,280,000	14,300 - 42,700	64 - 216
Long Stroke			360 - 480

The unit characteristics are used in the standard designation:

$$\underbrace{\text{C}}_{\text{type}} - \underbrace{\text{228 D}}_{\text{torque}} - \underbrace{\text{200}}_{\text{load}} - \underbrace{\text{74}}_{\text{stroke}}$$

Conventional, 228,000 in-lb Double gear reducer, 20,000 lb beam load, 74 in. maximum stroke.

Applications

Conventional: comprise the majority of applications.

Beam Balanced: generally shallow low flow wells.

Air Balanced: deep and high volume wells. Offshore wells.

Mark II: moderate to high volume wells.

Long Stroke: Viscous crude, deep wells.

b) Prime Mover: generally consists of an electric motor operating at approximately 1750 RPM. The desired pumping speed is obtained by selecting appropriate size V-belt pulleys in relation to the unit's gear reducer.

Natural gas internal combustion engines are also used generally in remote locations. Casinghead gas can be used as fuel.

c) Sucker Rods: Transmit reciprocating motion from the pumping unit to the subsurface pump. They are subjected to cyclic loading in a corrosive environment. Thus fatigue and corrosion are the principal constraints in design and selection of sucker rods.

Standard steel rods are manufactured in diameter from 1/2" to 1-^{1}/8" to cover the wide range of applications.

Tapered rod strings are commonly used in deep wells in order to optimize the utilization of rods and reduce overall loading. API RP11L presents recommended combinations of rod sizes as a function of the diameter of the pump plunger.

TABLE 4.1
ROD AND PUMP DATA
See Par. 4.5.

1	2	3	4	5	6	7	8	9	10	11
Rod* No.	Plung. Diam., inches D	Rod Weight, lb per ft W_r	Elastic Constant, in. per lb ft E_r	Frequency Factor, F_c	Rod String, % of each size 1⅛	1	⅞	¾	⅝	½
44	All	0.726	1.990×10^{-6}	1.000						100.0
54	1.06	0.908	1.668×10^{-6}	1.138					44.6	55.4
54	1.25	0.929	1.633×10^{-6}	1.140					49.5	50.5
54	1.50	0.957	1.584×10^{-6}	1.137					56.4	43.6
54	1.75	0.990	1.525×10^{-6}	1.122					64.6	35.4
54	2.00	1.027	1.460×10^{-6}	1.095					73.7	26.3
54	2.25	1.067	1.391×10^{-6}	1.061					83.4	16.6
54	2.50	1.108	1.318×10^{-6}	1.023					93.5	6.5
55	All	1.135	1.270×10^{-6}	1.000					100.0	
64	1.06	1.164	1.382×10^{-6}	1.229				33.3	33.1	33.6
64	1.25	1.211	1.319×10^{-6}	1.215				37.2	35.9	26.9
64	1.50	1.275	1.232×10^{-6}	1.184				42.3	40.4	17.3
64	1.75	1.341	1.141×10^{-6}	1.145				47.4	45.2	7.4
65	1.06	1.307	1.138×10^{-6}	1.098				34.4	65.6	
65	1.25	1.321	1.127×10^{-6}	1.104				37.3	62.7	
65	1.50	1.343	1.110×10^{-6}	1.110				41.8	58.2	
65	1.75	1.369	1.090×10^{-6}	1.114				46.9	53.1	
65	2.00	1.394	1.070×10^{-6}	1.114				52.0	48.0	
65	2.25	1.426	1.045×10^{-6}	1.110				58.4	41.6	
65	2.50	1.460	1.018×10^{-6}	1.099				65.2	34.8	
65	2.75	1.497	0.990×10^{-6}	1.082				72.5	27.5	
65	3.25	1.574	0.930×10^{-6}	1.037				88.1	11.9	
66	All	1.634	0.883×10^{-6}	1.000				100.0		
75	1.06	1.566	0.997×10^{-6}	1.191			27.0	27.4	45.6	
75	1.25	1.604	0.973×10^{-6}	1.193			29.4	29.8	40.8	
75	1.50	1.664	0.935×10^{-6}	1.189			33.3	33.3	33.3	
75	1.75	1.732	0.892×10^{-6}	1.174			37.8	37.0	25.1	
75	2.00	1.803	0.847×10^{-6}	1.151			42.4	41.3	16.3	
75	2.25	1.875	0.801×10^{-6}	1.121			46.9	45.8	7.2	
76	1.06	1.802	0.816×10^{-6}	1.072			28.5	71.5		
76	1.25	1.814	0.812×10^{-6}	1.077			30.6	69.4		
76	1.50	1.833	0.804×10^{-6}	1.082			33.8	66.2		
76	1.75	1.855	0.795×10^{-6}	1.088			37.5	62.5		
76	2.00	1.880	0.785×10^{-6}	1.093			41.7	58.3		
76	2.25	1.908	0.774×10^{-6}	1.096			46.5	53.5		
76	2.50	1.934	0.764×10^{-6}	1.097			50.8	49.2		
76	2.75	1.967	0.751×10^{-6}	1.094			56.5	43.5		
76	3.25	2.039	0.722×10^{-6}	1.078			68.7	31.3		
76	3.75	2.119	0.690×10^{-6}	1.047			82.3	17.7		
77	All	2.224	0.649×10^{-6}	1.000			100.0			
85	1.06	1.883	0.873×10^{-6}	1.261		22.2	22.4	22.4	33.0	
85	1.25	1.943	0.841×10^{-6}	1.253		23.9	24.2	24.3	27.6	
85	1.50	2.039	0.791×10^{-6}	1.232		26.7	27.4	26.8	19.2	
85	1.75	2.138	0.738×10^{-6}	1.201		29.6	30.4	29.5	10.5	

TABLE 4.1 (Continued)
See Par. 4.5.

1	2	3	4	5	6	7	8	9	10	11
Rod* No.	Plunger Diam., inches D	Rod Weight, lb per ft W_r	Elastic Constant, in. per lb ft E_r	Frequency Factor, F_c	Rod String, % of each size 1 1/8	1	7/8	3/4	5/8	1/2
86	1.06	2.058	0.742×10^{-6}	1.151		22.6	23.0	54.3		
86	1.25	2.087	0.732×10^{-6}	1.156		24.3	24.5	51.2		
86	1.50	2.133	0.717×10^{-6}	1.162		26.8	27.0	46.3		
86	1.75	2.185	0.699×10^{-6}	1.164		29.4	30.0	40.6		
86	2.00	2.247	0.679×10^{-6}	1.161		32.8	33.2	33.9		
86	2.25	2.315	0.656×10^{-6}	1.153		36.9	36.0	27.1		
86	2.50	2.386	0.633×10^{-6}	1.138		40.6	39.7	19.7		
86	2.75	2.455	0.610×10^{-6}	1.119		44.5	43.3	12.2		
87	1.06	2.390	0.612×10^{-6}	1.055		24.3	75.7			
87	1.25	2.399	0.610×10^{-6}	1.058		25.7	74.3			
87	1.50	2.413	0.607×10^{-6}	1.062		27.7	72.3			
87	1.75	2.430	0.603×10^{-6}	1.066		30.3	69.7			
87	2.00	2.450	0.598×10^{-6}	1.071		33.2	66.8			
87	2.25	2.472	0.594×10^{-6}	1.075		36.4	63.6			
87	2.50	2.496	0.588×10^{-6}	1.079		39.9	60.1			
87	2.75	2.523	0.582×10^{-6}	1.082		43.9	56.1			
87	3.25	2.575	0.570×10^{-6}	1.084		51.6	48.4			
87	3.75	2.641	0.556×10^{-6}	1.078		61.2	38.8			
87	4.75	2.793	0.522×10^{-6}	1.038		83.6	16.4			
88	All	2.904	0.497×10^{-6}	1.000		100.0				
96	1.06	2.382	0.670×10^{-6}	1.222	19.1	19.2	19.5	42.3		
96	1.25	2.435	0.655×10^{-6}	1.224	20.5	20.5	20.7	38.3		
96	1.50	2.511	0.633×10^{-6}	1.223	22.4	22.5	22.8	32.3		
96	1.75	2.607	0.606×10^{-6}	1.213	24.8	25.1	25.1	25.1		
96	2.00	2.703	0.578×10^{-6}	1.196	27.1	27.9	27.4	17.6		
96	2.25	2.806	0.549×10^{-6}	1.172	29.6	30.7	29.8	9.8		
97	1.06	2.645	0.568×10^{-6}	1.120	19.6	20.0	60.3			
97	1.25	2.670	0.563×10^{-6}	1.124	20.8	21.2	58.0			
97	1.50	2.707	0.556×10^{-6}	1.131	22.5	23.0	54.5			
97	1.75	2.751	0.548×10^{-6}	1.137	24.5	25.0	50.4			
97	2.00	2.801	0.538×10^{-6}	1.141	26.8	27.4	45.7			
97	2.25	2.856	0.528×10^{-6}	1.143	29.4	30.2	40.4			
97	2.50	2.921	0.515×10^{-6}	1.141	32.5	33.1	34.4			
97	2.75	2.989	0.503×10^{-6}	1.135	36.1	35.3	28.6			
97	3.25	3.132	0.475×10^{-6}	1.111	42.9	41.9	15.2			
98	1.06	3.068	0.475×10^{-6}	1.043	21.2	78.8				
98	1.25	3.076	0.474×10^{-6}	1.045	22.2	77.8				
98	1.50	3.089	0.472×10^{-6}	1.048	23.8	76.2				
98	1.75	3.103	0.470×10^{-6}	1.051	25.7	74.3				
98	2.00	3.118	0.468×10^{-6}	1.055	27.7	72.3				
98	2.25	3.137	0.465×10^{-6}	1.058	30.1	69.9				
98	2.50	3.157	0.463×10^{-6}	1.062	32.7	67.3				
98	2.75	3.180	0.460×10^{-6}	1.066	35.6	64.4				
98	3.25	3.231	0.453×10^{-6}	1.071	42.2	57.8				
98	3.75	3.289	0.445×10^{-6}	1.074	49.7	50.3				
98	4.75	3.412	0.428×10^{-6}	1.064	65.7	34.3				
99	All	3.676	0.393×10^{-6}	1.000	100.0					

TABLE 4.1 (Continued)
See Par. 4.5.

1	2	3	4	5	6	7	8	9	10	11
Rod* No.	Plunger Diam., inches D	Rod Weight, lb per ft W_r	Elastic Constant, in. per lb ft E_r	Frequency Factor, F_c	Rod String, % of each size 1¼	1⅛	1	⅞	¾	⅝
107	1.06	2.977	0.524 x 10^{-6}	1.184	16.9	16.8	17.1	49.1	......	......
107	1.25	3.019	0.517 x 10^{-6}	1.189	17.9	17.8	18.0	46.3	......	......
107	1.50	3.086	0.506 x 10^{-6}	1.196	19.4	19.2	19.5	41.9	......	......
107	1.75	3.158	0.494 x 10^{-6}	1.197	21.0	21.0	21.2	36.9	......	......
107	2.00	3.238	0.480 x 10^{-6}	1.195	22.7	22.8	23.1	31.4	......	......
107	2.25	3.336	0.464 x 10^{-6}	1.187	25.0	25.0	25.0	25.0	......	......
107	2.50	3.435	0.447 x 10^{-6}	1.174	26.9	27.7	27.1	18.2	......	......
107	2.75	3.537	0.430 x 10^{-6}	1.166	29.1	30.2	29.3	11.3	......	......
108	1.06	3.325	0.447 x 10^{-6}	1.097	17.3	17.8	64.9	......	......	......
108	1.25	3.345	0.445 x 10^{-6}	1.101	18.1	18.6	63.2	......	......	......
108	1.50	3.376	0.441 x 10^{-6}	1.106	19.4	19.9	60.7	......	......	......
108	1.75	3.411	0.437 x 10^{-6}	1.111	20.9	21.4	57.7	......	......	......
108	2.00	3.452	0.432 x 10^{-6}	1.117	22.6	23.0	54.3	......	......	......
108	2.25	3.498	0.427 x 10^{-6}	1.121	24.5	25.0	50.5	......	......	......
108	2.50	3.548	0.421 x 10^{-6}	1.124	26.5	27.2	46.3	......	......	......
108	2.75	3.603	0.415 x 10^{-6}	1.126	28.7	29.6	41.6	......	......	......
108	3.25	3.731	0.400 x 10^{-6}	1.123	34.6	33.9	31.6	......	......	......
108	3.75	3.873	0.383 x 10^{-6}	1.108	40.6	39.5	19.9	......	......	......
109	1.06	3.839	0.378 x 10^{-6}	1.035	18.9	81.1	......	......	......	......
109	1.25	3.845	0.378 x 10^{-6}	1.036	19.6	80.4	......	......	......	......
109	1.50	3.855	0.377 x 10^{-6}	1.038	20.7	79.3	......	......	......	......
109	1.75	3.867	0.376 x 10^{-6}	1.040	22.1	77.9	......	......	......	......
109	2.00	3.880	0.375 x 10^{-6}	1.043	23.7	76.3	......	......	......	......
109	2.25	3.896	0.374 x 10^{-6}	1.046	25.4	74.6	......	......	......	......
109	2.50	3.911	0.372 x 10^{-6}	1.048	27.2	72.8	......	......	......	......
109	2.75	3.930	0.371 x 10^{-6}	1.051	29.4	70.6	......	......	......	......
109	3.25	3.971	0.367 x 10^{-6}	1.057	34.2	65.8	......	......	......	......
109	3.75	4.020	0.363 x 10^{-6}	1.063	39.9	60.1	......	......	......	......
109	4.75	4.120	0.354 x 10^{-6}	1.066	51.5	48.5	......	......	......	......

*Rod No. shown in first column refers to the largest and smallest rod size in eighths of an inch. For example, Rod No. 76 is a two-way taper of 7/8 and 6/8 rods. Rod No. 85 is a four-way taper of 8/8, 7/8, 6/8, and 5/8 rods. Rod No. 109 is a two-way taper of 1¼ and 1⅛ rods. Rod No. 77 is a straight string of 7/8 rods, etc.

As the pump diameter increases the fluid load carried by the rods increases and the percentage of large rods in a tapered string increases accordingly.

Rod Number 65		Rod Sizes in Combination and Their %	
		$\frac{6}{8}$"	$\frac{5}{8}$"
Plunger diameter	1.06"	34.4%	65.6%
	1.50"	41.8%	58.2%
	2.75"	75.2%	27.5%

Characteristics of Sucker Rods

a) Rod weight (W_r): Weight in air of the combined rod sizes -

Rod 65 x 106 pump $W_r = 1.291$ lb/ft.

Rod 55 $W_r = 1.135$ lb/ft (Uniform diamter)

Rod 66 $W_r = 1.634$ lb/ft (Uniform diameter)

b) Elastic Constant

Stretch of unit length of rod (1 ft) per unit load applied (lb)

Er Er $\frac{\text{in}}{\text{lb-ft}}$

Rod 65 x 1.06 pump $Er = 1.150 \times 10^{-6} \frac{\text{in}}{\text{lb-ft}}$

K_r = Spring constant of a given length rod string ($\frac{\text{lbs}}{\text{in}}$) load in pounds to stretch the total length L by one inch.

84

$$K_r = \frac{1}{E_r x L}$$

$$\frac{1}{K_r} = E_r \times L$$ = elastic constant of rod string of length L ($\frac{in}{lb}$)

c) <u>Frequency Factor</u> F_c

Ratio of natural frequency of uniform diameter rod string to natural frequency of tapered rod string.

$$\frac{N_o'}{N_o} = F_c \qquad N_o = \frac{N_o'}{F_c}$$

N_o' = tapered string frequency

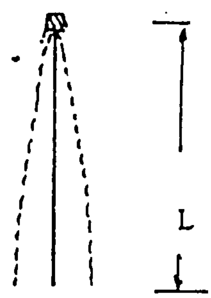
N_o = uniform diameter frequency (larger diameter)

Rod 65 x 1.06 pump -

F_c = 1.085

<u>Natural Frequency of Rod String</u>

Velocity of propagation in rod immersed in fluid = 16,300 ft/sec.



Fundamental mode. Maximum displacement (amplitude) at L.

$$F_o = \frac{C}{4L}$$

$$N_o = \frac{16300 \text{ ft/sec} \times 60 \text{ sec/min}}{4L} = \frac{245000}{L} \quad \left(\frac{\text{cycles}}{\text{minute}}\right)$$

Characteristics of Rod String

When sucker rods are combined into a rod string of a given length driving a given pump the characteristics of the system are expressed in terms of certain dimensionless parameters and in design calculations.

Rod Stretch

For a total force F applied to a rod string of length L the rod stretch will be given by:

$$S_r = F \times (E_r L) = \frac{F}{K_r}$$

The E_r factor can be calculated as $\frac{12}{A_r E}$

where A_r is section area in in^2

$E = 30 \times 10^6$ psi (Young's modulus)

For a tapered string:

$$S_r = \frac{12\,F}{E} \left[\frac{L_1}{A_1} \;\; \frac{L_2}{A_2} \;\; \frac{L_3}{A_3} \right]$$

E_r factors are tabulated in API RP11L for standard combinations.

Dimensionless Rod Stretch

For a unit operating with a polished rod stroke S, the ratio of the rod stretch to the stroke is a measure of the stiffness of the system and is defined as dimensionless rod stretch. When F_o (the static load correponding to the fluid load) is used as the force causing the rod stretch:

$$\text{Dimensionless rod stretch} = \frac{S_r}{S} = \frac{F_o}{SK_r}$$

This parameter is used in calculation of the pumping system performance.

Dimensionless Pumping Speed

This parameter expresses the relation between the speed of the pumping unit, N (strokes per minute) to the fundamental frequency of the rod string. (N_o or N_o')

$$\text{Dimensionless Pumping Speed:} \quad \frac{N}{N_o}$$

Generally asynchronous speeds ($N = \frac{2N_o}{3}, \frac{2N_o}{5}, \frac{2N_o}{7}$) are more desirable than synchronous speeds ($\frac{N_o}{2}, \frac{N_o}{4}, \ldots, \ldots$)

Maximum pumping speed corresponds to the asynchronous speed well below the free fall speed of the rods.

d) Downhole Pumps

Basically consist of positive displacement single cylinder pumps with one intake and one outlet valve. There are two basic types: tubing pumps and rod pumps.

Rod Pumps -- Run onto the sucker rods, they fit
into a seating nipple in the tubing.

Stationary barrel, top anchor

Stationary barrel, bottom anchor

Traveling barrel, bottom anchor.

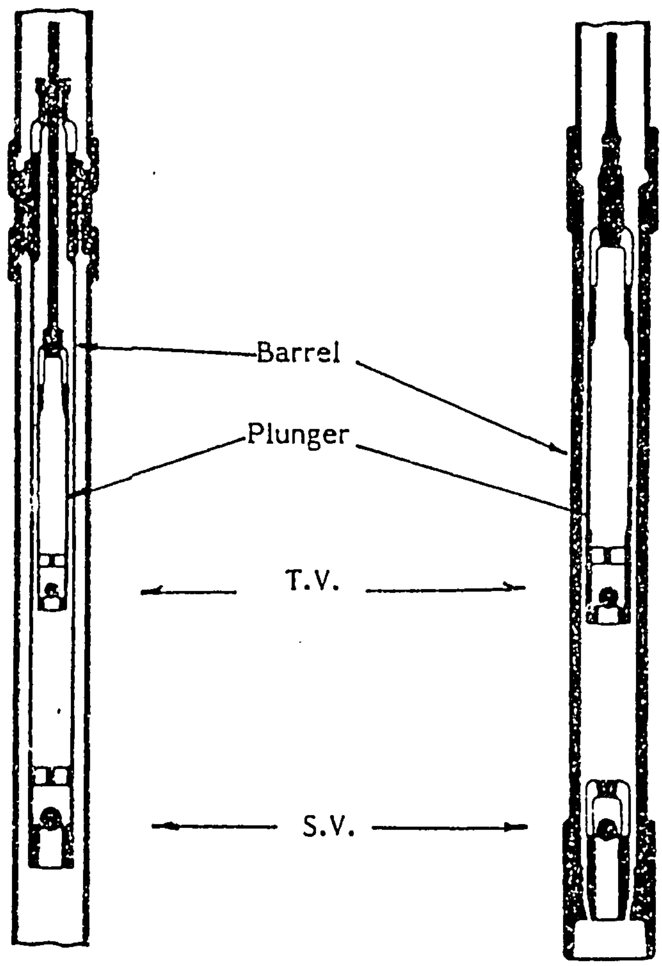
Tubing Pumps -- Barrel is run as a portion of the tubing.

Types of Barrels:

L - liner
H - heavy wall
metal plunger.

S - thin wall
P - heavy wall
soft packed

The following schematic diagram illustrates the two basic types.



Rod Pump

Tubing Pump

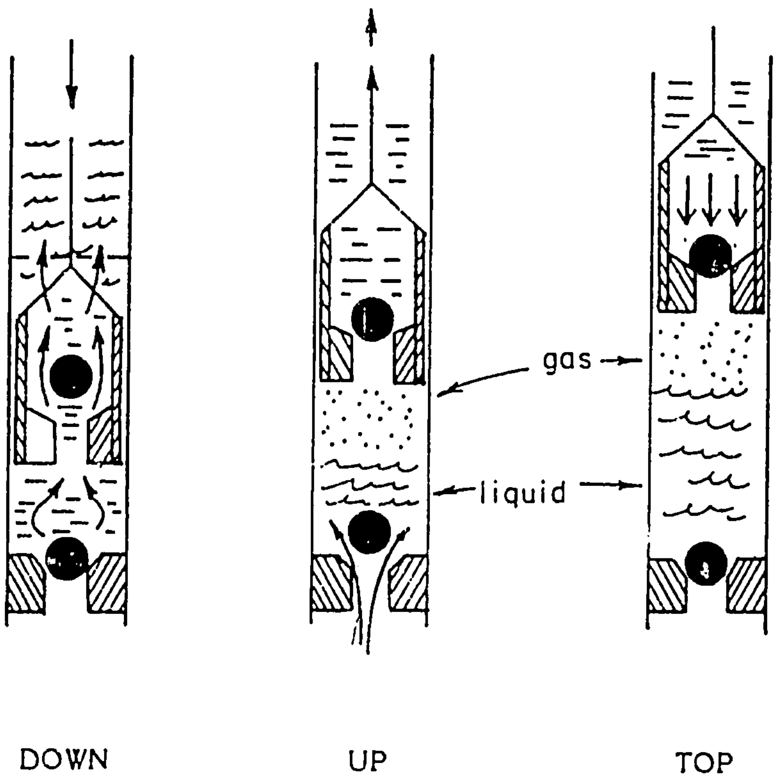
The rod pump offers the advantage of complete replacement without having to pull the tubing from the well. In the tubing pump both valves and plunger can be replaced by pulling the rods.

Tubing pumps offer the largest plunger area and thus the largest volumetric displacement.

Pumping Cycle

During one complete cycle fluid is admitted into the pump barrel and then transferred to the tubing through the hollow pump plunger. During the upstroke the fluid column in the tubing is supported by the sucker rods through the travelling valve. During the downstroke the fluid is supported by the tubing through the standing valve.

The following diagram illustrates the cycle for a tubing pump



Pump Capacity:

It is a function of the net stroke of the plunger, the plunger diameter, the pumping speed and the volumetric efficiency:

$$Q = 0.15 \times Ev \times Ap \times Sp \times N$$

Ev = volumetric efficiency

Sp = net plunger stroke (in)

Ap = plunger area (in^2)

N = pumping speed (SPM)

Standard pumps are characterized by a pump constant:

$$\text{Pump Constant} = \frac{\text{Barrels per day}}{\text{(SPM)(inch of travel)}}$$

so that the pump capacity can be expressed as:

$$Q = (\text{Pump Constant})\ (N)\ (Sp)$$

Also a fluid load factor is tabulated, which corresponds to the weight per foot of the fluid column supported by a given plunger diameter assuming a fluid specific gravity of 1.00.

The following tables present this information for standard size pumps.

PUMP CONSTANTS

1	2	3	4
Plunger diameter, in D_p	Plgr diam squared sq in D_p^2	Fluid load factor* lb per ft $(0.340 \cdot D_p^2)$	K Pump factor $(0.1166 \cdot D_p^2)$
1 1/16	1.1289	0.384	0.132
1 1/4	1.5625	0.531	0.182
1 1/2	2.2500	0.765	0.262
1 3/4	3.0625	1.041	0.357
2	4.0000	1.360	0.466
2 1/4	5.0625	1.721	0.590
2 1/2	6.2500	2.125	0.728
2 3/4	7.5625	2.571	0.881
3 3/4	14.0625	4.781	1.640
4 3/4	22.5625	7.671	2.630

*For fluids with specific gravity of 1.00.

TUBING DATA

1	2	3	4	5
Tubing size	Outside diameter, in.	Inside diameter, in.	Metal area, sq. in	Elastic constant, in. per lb ft E_t
1.900	1.900	1.610	0.800	0.500×10^{-6}
2 3/8	2.375	1.995	1.304	0.307×10^{-6}
2 7/8	2.875	2.441	1.812	0.221×10^{-6}
3 1/2	3.500	2.992	2.590	0.154×10^{-6}
4	4.000	3.476	3.077	0.130×10^{-6}
4 1/2	4.500	3.958	3.601	0.111×10^{-6}

MAXIMUM PUMP SIZE AND TYPE

Pump type	Tubing size, in.			
	1.900	2 3/8	2 7/8	3 1/2
Tubing one-piece, thin-wall barrel (TW)	1 1/2	1 3/4	2 1/4	2 3/4
Tubing one-piece, heavy-wall barrel (TH)	1 1/2	1 3/4	2 1/4	2 3/4
Tubing liner barrel (TL)	—	1 3/4	2 1/4	2 3/4
Rod one-piece, thin-wall barrel (RW)	1 1/4	1 1/2	2	2 1/2
Rod one-piece, heavy wall barrel (RH)	1 1/16	1 1/4	1 3/4	2 1/4
Rod liner barrel (RL)	—	1 1/4	1 3/4	2 1/4

SUCKER ROD DATA

1	2	3	4
Rod Size	Metal Area, Sq in.	Rod Weight in air, lb per ft W_r	Elastic Constant, in. per lb ft E_r
½	0.196	0.72	1.990×10^{-6}
⅝	0.307	1.13	1.270×10^{-6}
¾	0.442	1.63	0.883×10^{-6}
⅞	0.601	2.22	0.649×10^{-6}
1	0.785	2.90	0.497×10^{-6}
1⅛	0.994	3.67	0.393×10^{-6}

Pump Plunger Sizes Recommended for Optimum Conditions

Net Lift of Fluid ft.	Fluid Production - Barrels per day - 80 pct efficiency									
	100	200	300	400	500	600	700	800	900	1000
2000	1 1/2 1 1/4	1 3/4 1 1/2	2 1 3/4	2 1/4 2	2 1/2 2 1/4	2 3/4 2 1/2	2 3/4	2 3/4	2 3/4	2 3/4
3000	1 1/2 1 1/4	1 3/4 1 1/2	2 1 3/4	2 1/4 2	2 1/2 2 1/4	2 1/2 2 1/4	2 3/4 2 1/2	2 3/4	2 3/4	2 3/4
4000	1 1/4	1 3/4 1 1/2	2 1 3/4	2 1/4 2	2 1/4 2	2 1/4	2 1/4	2 1/4		
5000	1 1/4	1 3/4 1 1/2	2 1 3/4	2 1 3/4	2 1/4 2	2 1/4				
6000	1 1/4	1 1/2 1 1/4	1 3/4 1 1/2	1 3/4						
7000	1 1/4 1 1/8	1 1/2 1 1/4								
8000	1 1/4 1 1/8									

In this tabulation surface pumping strokes up to 74 in. only are considered.

Pump Efficiency

The volumetric efficiency of the pump depends principally upon the fillage of the barrel. This in turn is related to the presence of gas, the viscosity of the fluid being pumped and the pumping speed.

Slippage of oil past the plunger also affects the volumetric efficiency.

Slippage of Oil Past Plunger

$$Q/_{leak} = \frac{3.6 \times d \times c^3 \times \Delta P \times 10^6}{v \times L}$$

d = plunger diameter (inch)

c = clearance (inch)

P = differential pressure across plunger (psi)

v = kinematic viscosity (centistokes)

L = plunger length (inches)

In addition leakage of fluid past the valves reduces the pump efficiency requiring valve replacement.

Net Plunger Stroke

The cyclic nature of the rod and pump motion introduces sequential loading and unloading of the sucker rods and tubing. The calculation of plunger stroke thus requires correcting the surface stroke at the polished rod for the elastic deformation (stretch) of the rods and tubing. The dynamic nature of the problem requires consideration of acceleration and vibration effects due to forcing oscillation of the rod string. The generalized problem including stress wave propagation has been solved numerically for numerous cases of standard combinations of rod sizes and pumps resulting in generalized dimensionless plots presented in API RPllL.

In general terms

$$Sp = S - Sr - St + \text{overtravel}$$

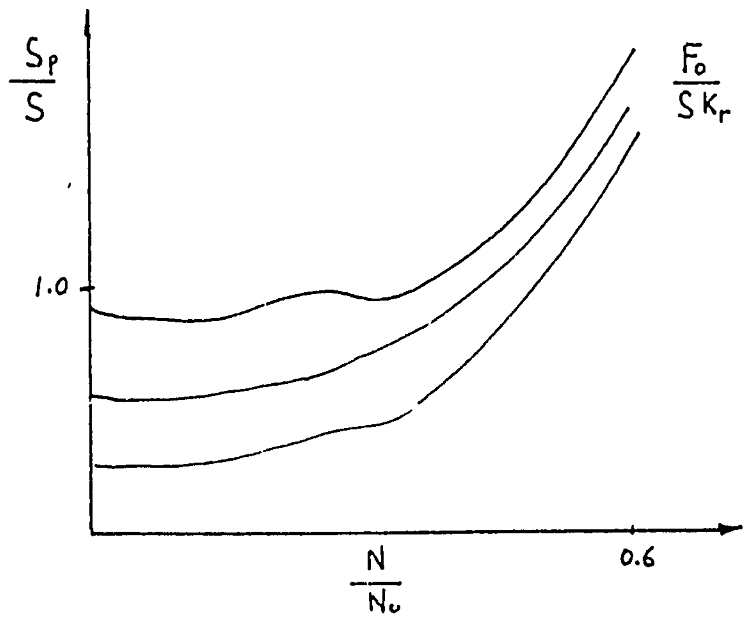
Sr = rod stretch (inches) due to fluid load

St = tubing stretch due to fluid load (= zero if tubing is anchored)

overtravel = acceleration and dynamic effects

Dimensionless Plunger Travel

$\frac{Sp}{S}$ is a function of pumping speed N and rod elasticity $\frac{F_o}{Sk_r}$



Therefore for a given rod $(\frac{F_o}{Sk_r})$ and $(\frac{N}{N_o})$ the value $(\frac{Sp}{S})$ is known.

$Sp = (\frac{Sp}{S}) \times S$ for anchored tubing

If tubing is not anchored:

$$Sp = (\frac{Sp}{S}) \times S - \frac{F_o}{K_t}$$

where K_t is defined as:

$\frac{1}{K_t} = E_t L$ where E_t = tubing Elastic Constant $\frac{in}{lb\text{-}ft}$

$$E_t = \frac{12}{A_t E}$$

and is tabulated in Table 2.4

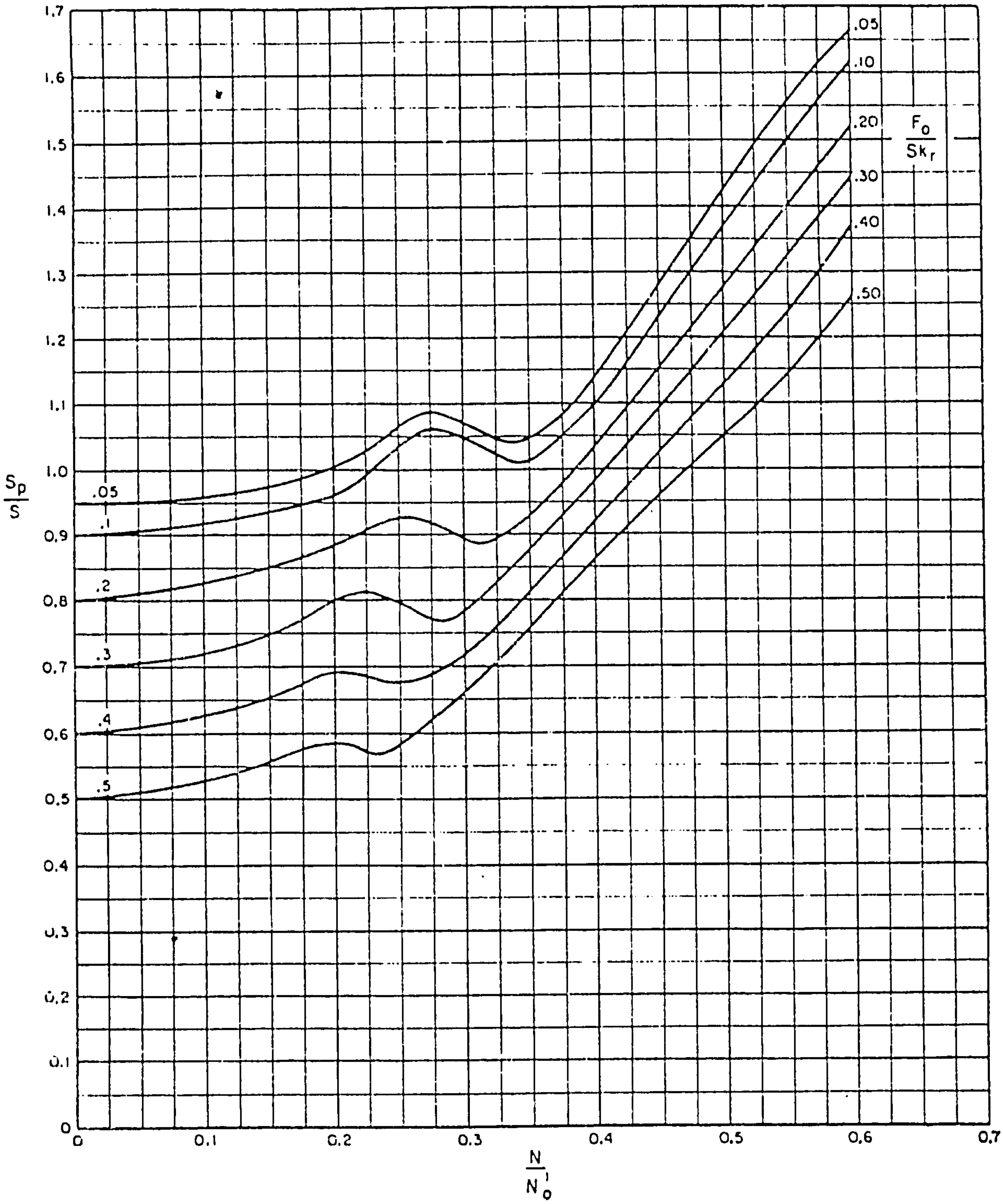


FIGURE 4.1

$\frac{S_p}{S}$, PLUNGER STROKE FACTOR

If the above dimensionless plot is not available the net plunger travel can be calculated as:

$$Sp = S - \frac{12\,F_o}{E}\left[\frac{L_1}{A_1} + \frac{L_2}{A_2} + \frac{L_3}{A_3}\right] - \frac{12\,F_o}{E}\quad\frac{L_T}{A_t}$$

$$+ \frac{12}{E}\,W_r\left(\frac{SN^2M}{70500}\right)\left[\frac{L_1}{A_1} + \frac{L_2}{A_2} + \frac{L_3}{A_3}\right]$$

The above relation does not include dynamic effects caused by stress wave propagation

L_1, L_2, L_3 = lengths of rods of areas A_1, A_2, A_3

L_T = tubing length, (ft)

A_t = tubing cross-sectional area (in^2)

W_r = average rod weight (lb/ft)

S = surface stroke (in)

N = pumping speed (SPM)

E = Young's modulus (psi)

M = machinery factor accounting for pumping unit geometry and its effect on acceleration.

Calculation of Loads

Pumping system design requires the calculation of maximum and minimum loads experienced during the pump cycle. These loads determine the size of the pumping unit the torque rating of the gear reducer and the power of the prime mover.

Peak Polished Rod Load =

Weight of rods in fluid + fluid load + acceleration + friction d Rod Load =

t of rods in fluid - acceleration - friction

re determined from static loads by considering acceleration

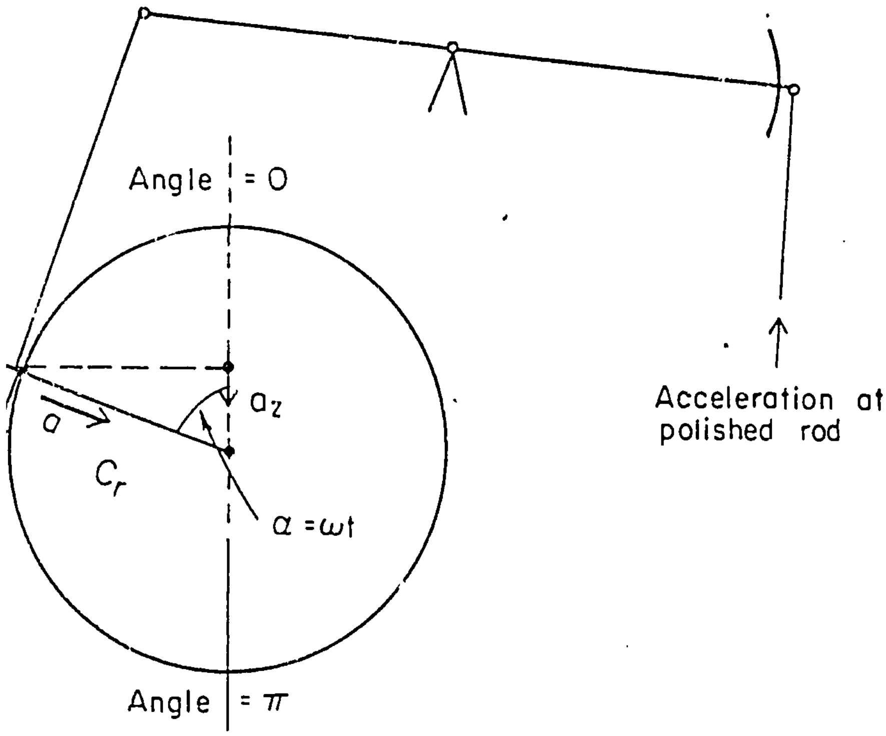
ects.

tor

n is simplified considering that sucker rods are subjected to

motion.

ollowing schematic diagram for a conventional unit.



where

C_r = crank radius (in)

$= \frac{2\pi N}{60}$ radians/sec.

N = strokes per minute

v = velocity in/sec.

the vertical component of acceleration:

$$a_z = a \cos \omega t \quad \text{so that}$$

the maximum and minimum accelerations correspond to $\omega t = o$ and $\omega t = \pi$ so that

$$a_{max} = \left(\frac{2\pi N}{60} \right)^2 C_r = \frac{N^2 C_r}{91.2}$$

The acceleration at the polished rod is obtained by introducing a machinery factor (M) dependent on the geometry of the unit and setting $S = 2 C_r/M$ = polished rod stroke

$$a_r = M \frac{N^2 S}{91.2}$$

The dynamic load at the rods is given by

$$W_{r/dynamic} = W_{r/static} \left(1 \pm \frac{a_r}{g} \right)$$

where $\frac{a_r}{g}$ = acceleration factor = C

Then $C = \frac{MN^2S}{70500}$ (Dimensionless for S inches and N, Spm).

In practice, acceleration factor should be less than 0.3 so that a maximum pumping speed can be estimated as:

$$N_{max} = \sqrt{\frac{21150}{(S)(M)}}$$

The acceleration factor does not include other effects due to vibration and stress wave propagation in the sucker rods.

Dimensionless Load Analysis

API RP11L provides dimensionless charts that include acceleration and vibration effects. These will depend on how near resonance frequency the pumping speed is, and the elasticity of the rod string. They are expressed in dimensionless form relative to Sk_r (force that causes an elongation equal to S)

$\frac{F_1}{Sk_r}$, dimensionless peak polished rod load factor

$\frac{F_2}{Sk_r}$, dimensionless minimum polished rod load factor

with the rod string being characterized by $\frac{F_o}{Sk_r}$ where F_o = static fluid load (lbs) defined as dimensionless rod stretch

$$\frac{\;\frac{F_o}{K_r}\;}{S}\quad \begin{array}{l}\text{(stretch caused by fluid load)}\\ \text{(Polished rod stroke)}\end{array}$$

For the maximum load charts have been prepared as follows:

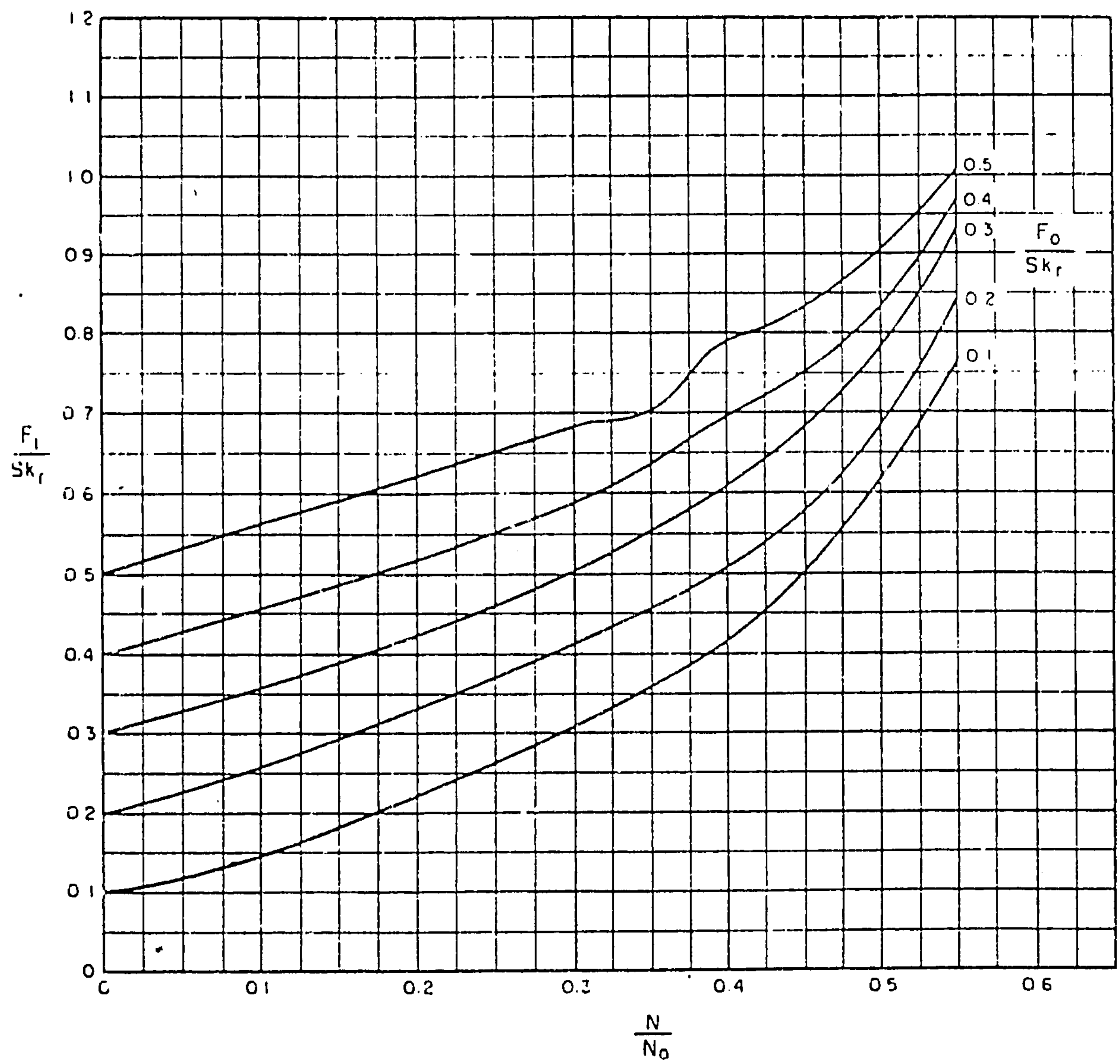


FIGURE 4.2

$\frac{F_1}{Sk_r}$, PEAK POLISHED ROD LOAD

For a given rod and pumping speed the $\frac{N}{N_o}$ and $\frac{F_o}{Sk_r}$ parameters are calculated and the value of $\left(\frac{F_1}{Sk_r} \right)$ is obtained from the chart. The peak polished rod load is then:

$$PPRL = Wr_f + \left(\frac{F_1}{Sk_r} \right) \times Sk_r$$

where Wr_f is the weight of rods in fluid.

ĮÕV

For the minimum load:

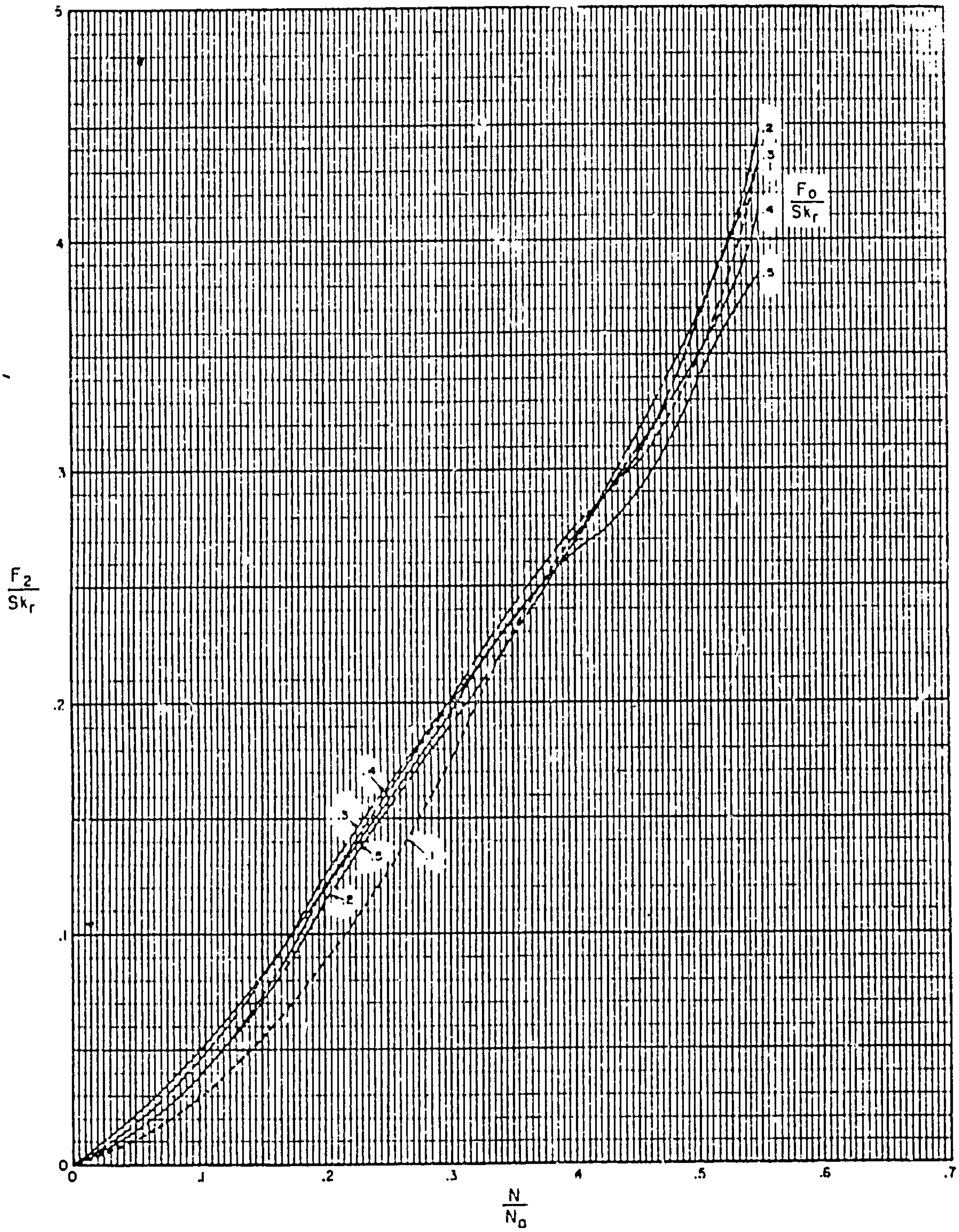


FIGURE 4.3

$\frac{F_2}{Sk_r}$, MINIMUM POLISHED ROD LOAD

In the same way

$$MPRL = Wr_f - \left(\frac{F_2}{Sk_r}\right) \times Sk_r$$

Torque Calculation

Peak torque determines the size of the gear reducer for a given unit.

Torque is a function of pitman pull, length of stroke and angular position of crank.

$$T_{max} = (W_{max} - W_B) \frac{S}{2} \times T_d \quad \text{in-lb}$$

W_{max} = peak polished rod load (PPRL) (lbs)

W_B = effective counterbalance load on polished rod (lbs)

S = stroke (inches)

T_d = dynamic factor

Counterbalance

Even distribution of loads - - even load on upstroke and downstroke - - Allows use of smaller transmission and prime recover

Upstroke

At crank:

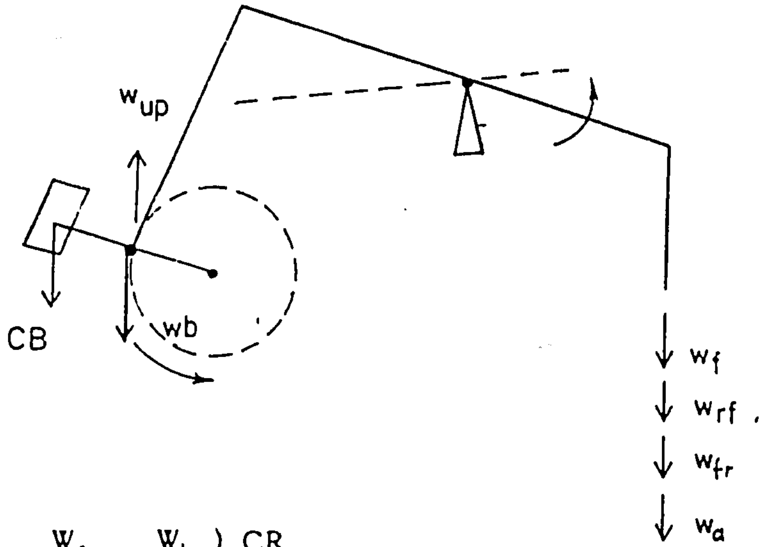
W_f ↑

W_{rf} ↑

W_a ↑

W_{fr} ↑

W_b ↓



$$\text{Torque} = (W_f + W_{rf} + W_a \quad W_{fr} \quad W_b)\ CR$$

Downstroke

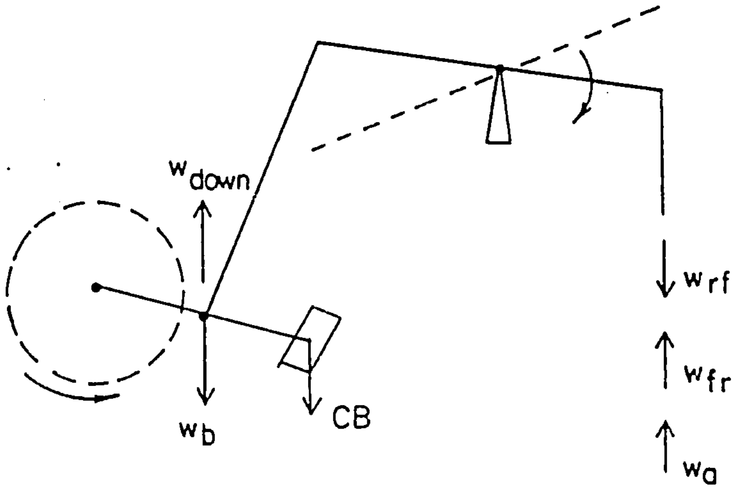
At crank:

W_{rf} ↑

W_{fr} ↓

W_a ↓

W_b ↓



$$\text{Torque} = (W_b - W_{rf} + W_a + W_{fr})\ CR$$

For ideal operating conditions set:

Torque Up = Torque Down

and assume same acceleration and friction

$$W_f + 2\,Wr_f = 2\,W_b$$

$$W_b = Wr_f + \frac{1}{2}\,W_f$$

<u>NOTE</u>: W_b is the counterbalance effect at the crank

$\approx$ counterbalance effect at the P.R.

$$Wr_f = W_r - W_{rb} = \frac{7.85 - 1}{7.85}\,W_r = 0.87\,W_r$$

7.85 = specific gravity of steel

1 = specific gravity of water

<u>Ideal Counterbalance</u>

$$W_b = 0.87\,W_r + \frac{1}{2}\,W_f$$

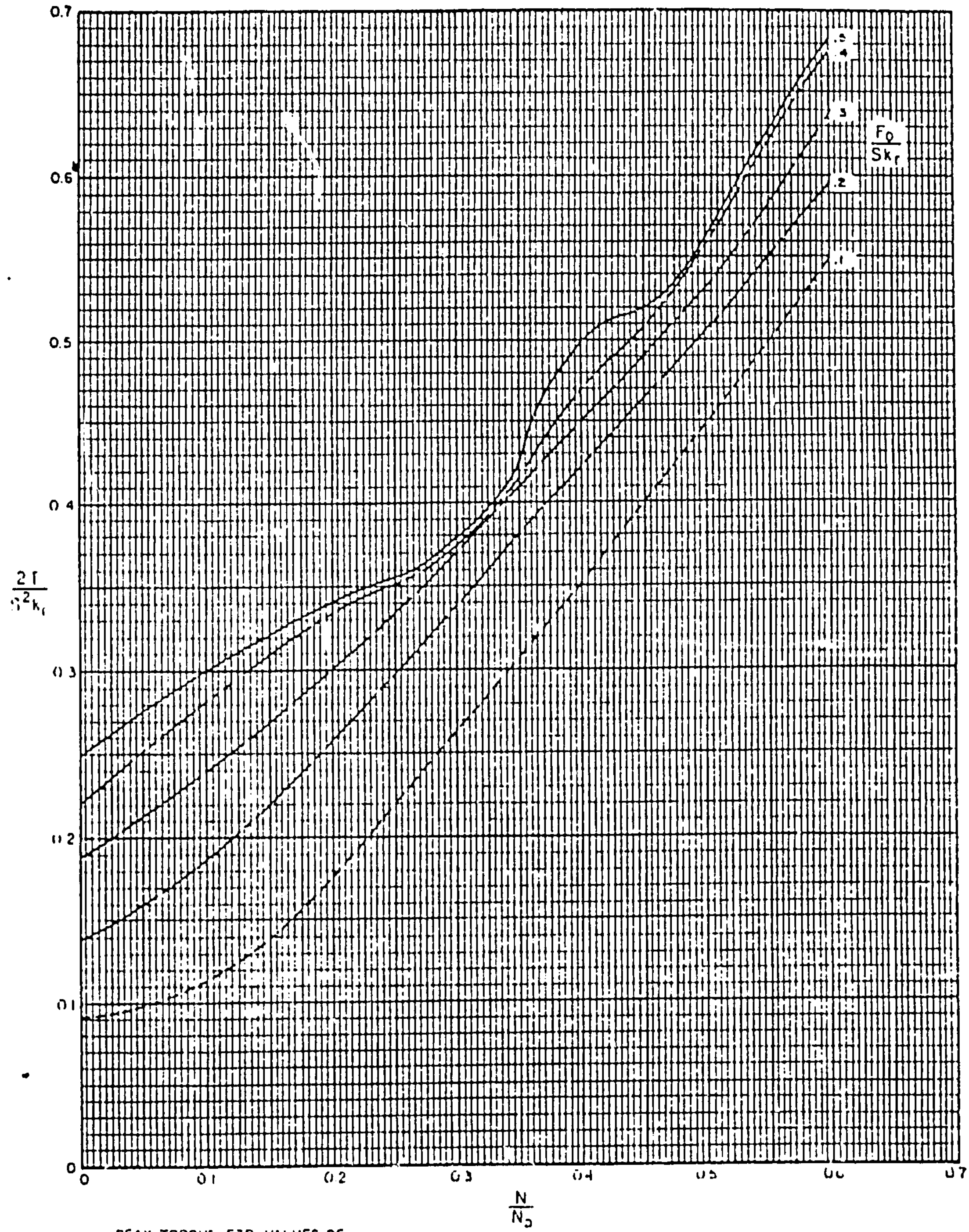
The above assumes that kinetic and frictional effects are the same in the upstroke and downstroke. This is not true due to the unit's geometry.

<u>Effect of Rod Dynamics</u>

Peak torque also is a function of the rod dynamics - since it depends on the peak load. This is handled with dimensionless plots of torque:

$$PT = \left(\frac{2T}{S^2K_r}\right) \times Sk_r \times \frac{S}{2} \times Ta$$

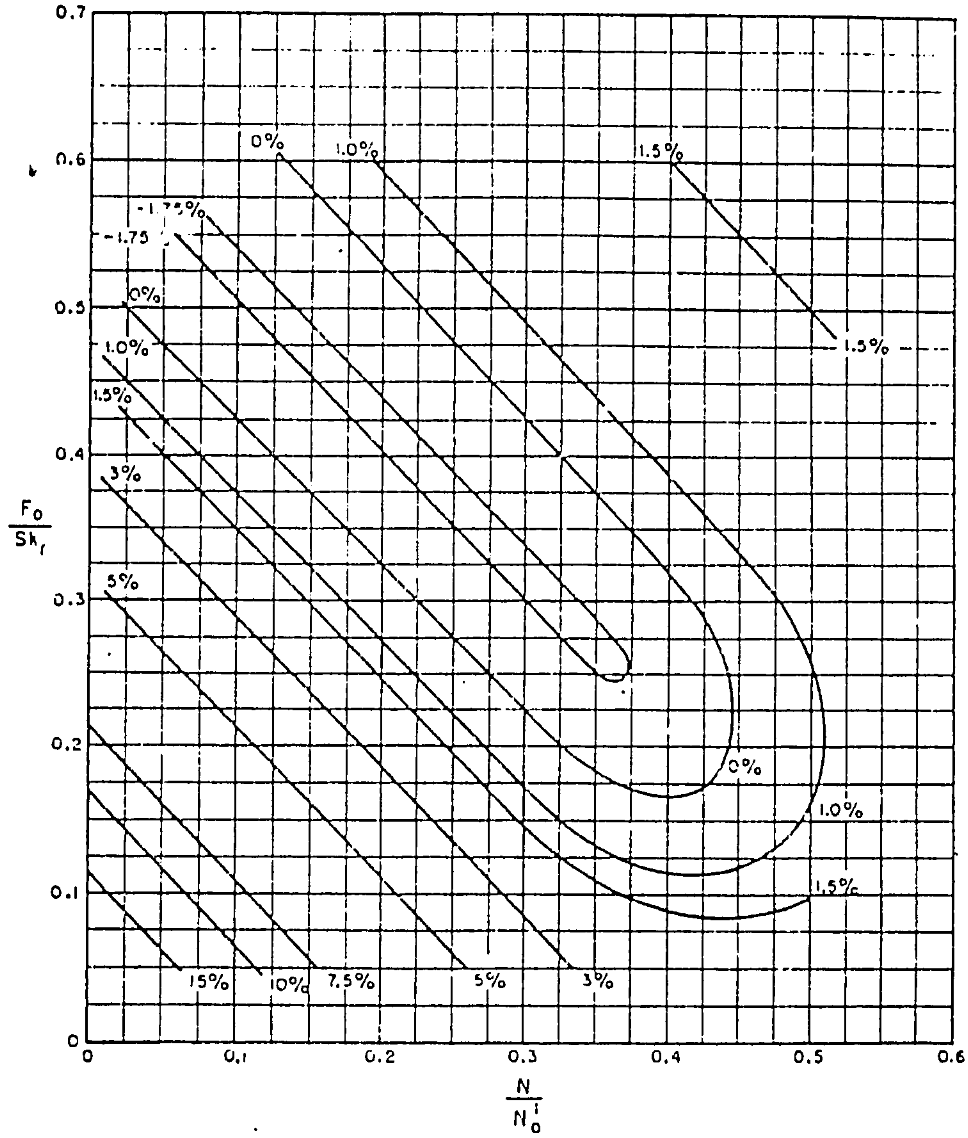
Ta is adjustment necessary if $\frac{Wr_f}{Sk_r} \neq 0.3$



PEAK TORQUE FOR VALUES OF W_H/Sk_r = 3 USE TORQUE ADJUSTMENT FOR OTHER VALUES OF W_H/Sk_r.

FIGURE 4.4

$\frac{2T}{S^2k_r}$, PEAK TORQUE



TO USE: MULTIPLY % INDICATED ON CURVE BY $\dfrac{\left(\frac{W_{rf}}{Sk_r} - 0.3\right)}{0.1}$

FOR EXAMPLE: $\frac{W_{rf}}{Sk_r} = 0.600$

$\frac{N}{N_o'} = 0.200 \qquad \frac{F_o}{Sk_r} = 0.188$

ADJUSTMENT = 3% FOR EACH 0.1 INCREASE IN $\frac{W_{rf}}{Sk_r}$ ABOVE 0.3

TOTAL ADJUSTMENT = 3 x 3% = 9%

T_a = 1.00 + 0.09 = 1.09

NOTE: IF $\frac{W_{rf}}{Sk_r}$ IS LESS THAN 0.3 ADJUSTMENT BECOMES NEGATIVE

FIGURE 4.6

T_a, ADJUSTMENT FOR PEAK TORQUE FOR VALUES OF $\frac{W_{rf}}{Sk_r}$ OTHER THAN 0.3

Power Requirements

Prime mover horsepower can be estimated from the hydraulic power required to lift fluid to the surface:

Hydraulic Horsepower = (0.00737) (D) (Q)

D = net lift in thousands of feet

Q = flow rate BBl/day

The brake horsepower is then estimated applying an efficiency factor.

$$\text{Brake Horsepower} = \frac{\text{Hydraulic Horsepower}}{\text{efficiency}}$$

Efficiency 0.35

A more accurate calculation of power can be undertaken using the dimensionless plot of peak polished rod horsepower from API RP11L.

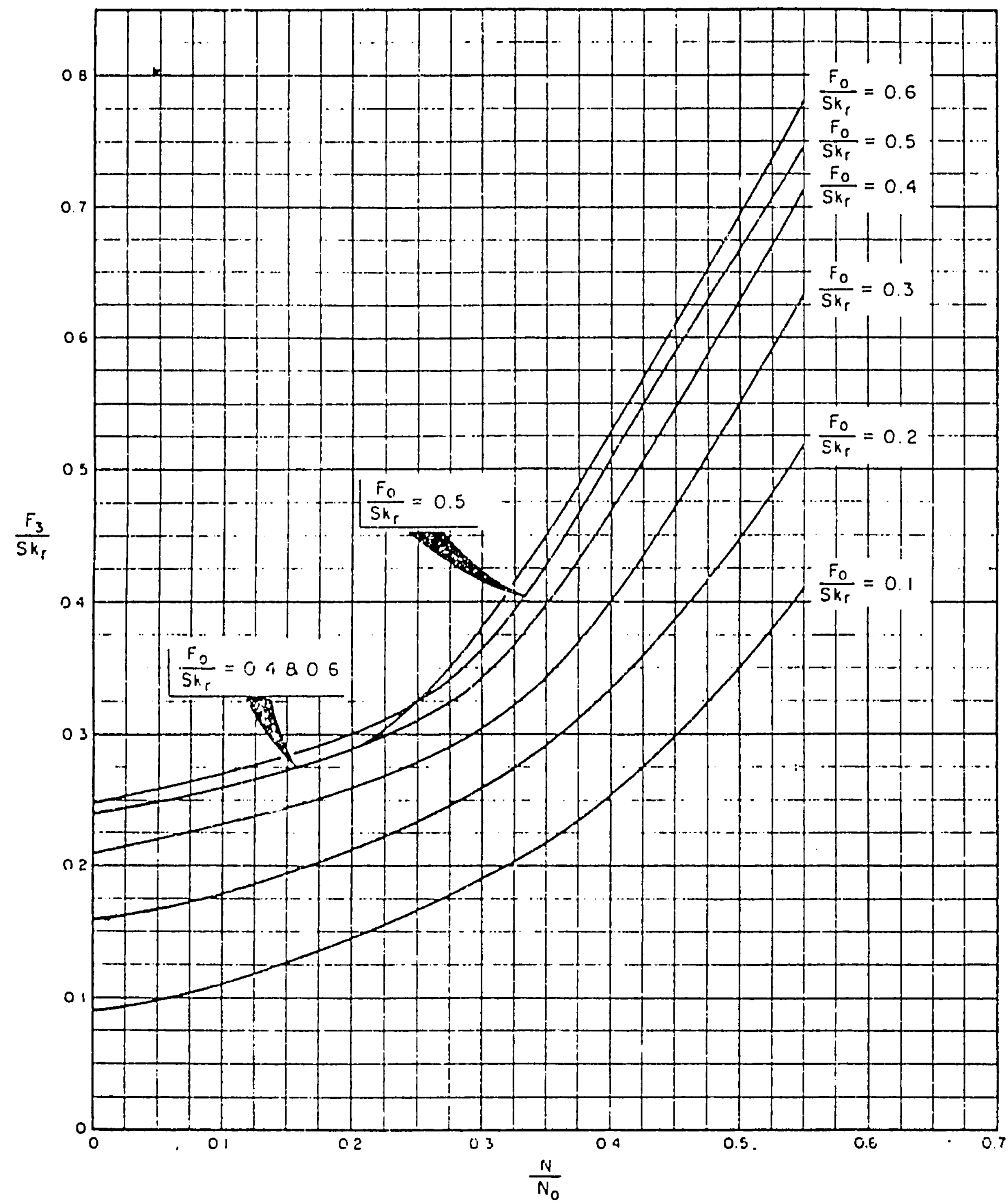
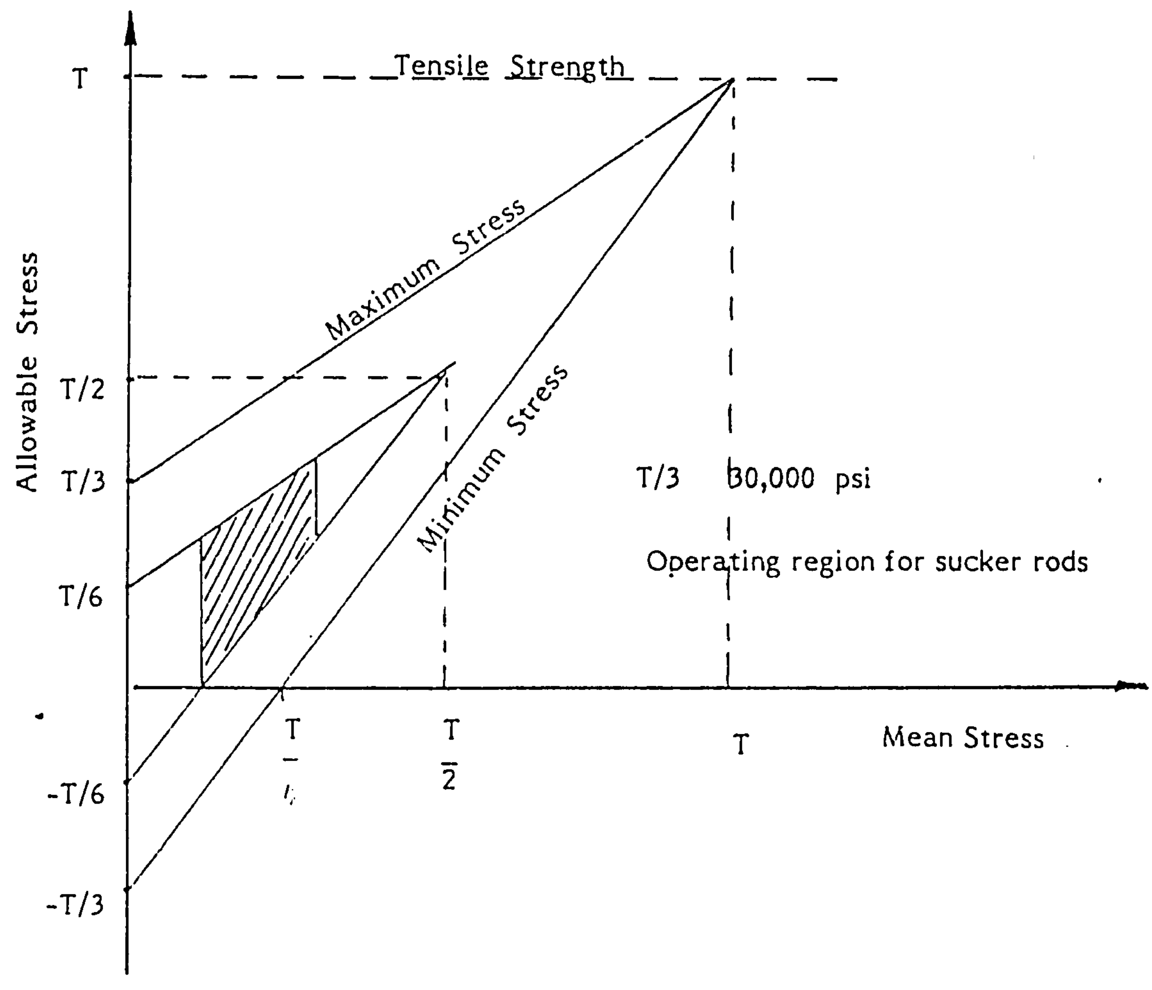


FIGURE 4.5

$\frac{F_3}{Sk_r}$, POLISHED ROD HORSE POWER

Rod String Fatigue

A modified Goodman diagram is used to insure that the maximum and minimum rod loads result in rod stresses within acceptable range.



For sucker rods the operating region is defined by:

$$T_{max} = \frac{1}{6}(T + 4\,T_{av})$$

provided

$$T_{max} \leq T/3$$

and the corresponding allowable minimum stress:

$$T_{min} = \frac{1}{6} (8\ T_{av} - T)$$

provided

$$T_{min} \geq o$$

T_{max} and T_{min} are obtained by dividing PPRL and MPRL by the rod's cross-sectional area.

Pumping System Design

Whether the design involves selecting a new unit or using an existing unit the objective will be to produce the desired amount of fluids without exceeding the:

Allowable Beam Load

Allowable Torque

Rod Stress

The basic information required includes:

Formation Productivity

Fluid Properties

The productivity is used to calculate the depth of the working fluid level at the desired rate. The fluid properties are used in the load calculations and to determine special restrictions on pumping speed (high viscosity).

Initial choices are made for:

Pump depth

Pumping speed

Polished rod stroke

Pump diameter

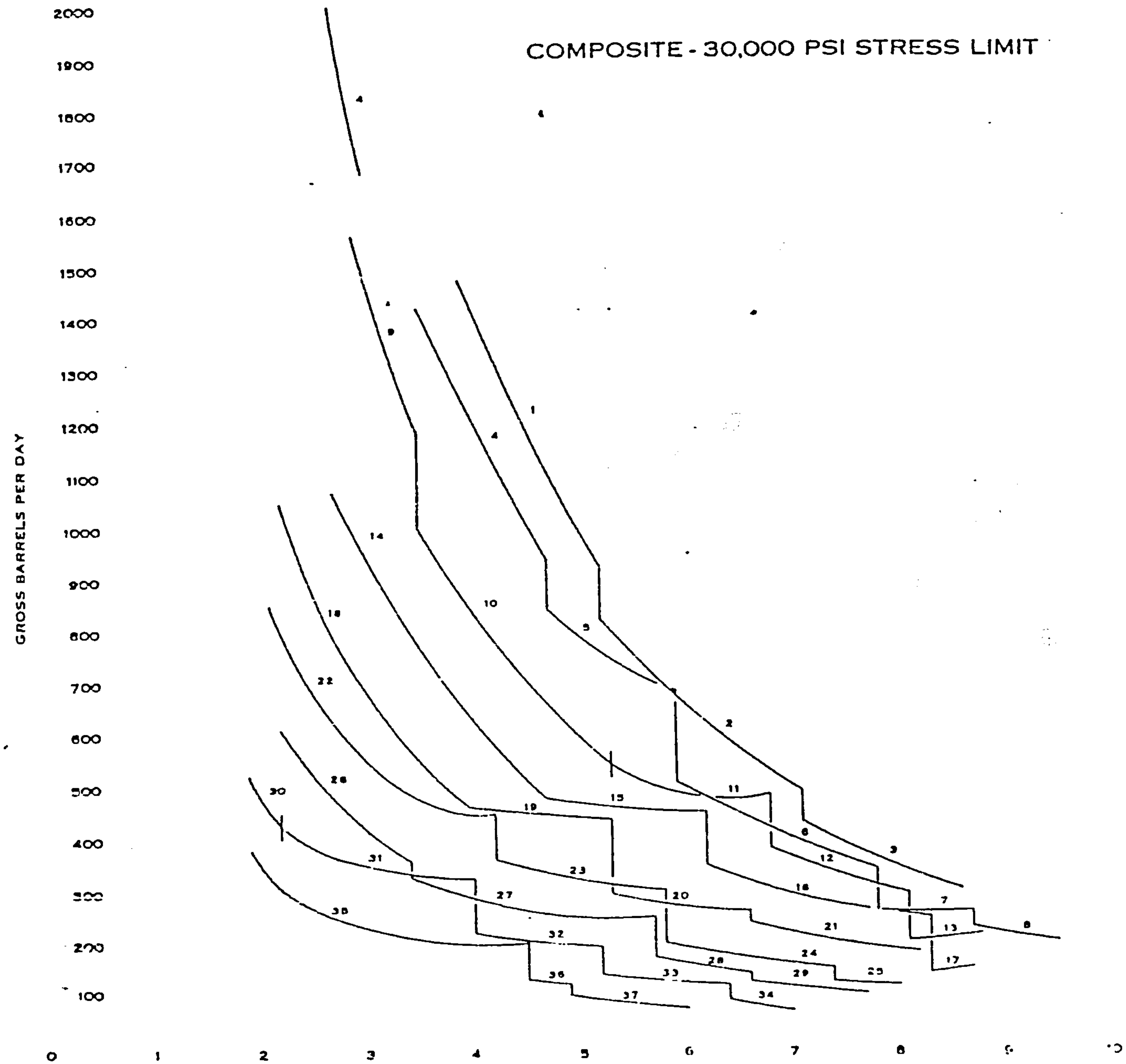
Rod string design.

Then the plunger stroke and pump displacement are calculated and checked with the desired rate. If not satisfied new parameters (rod size, pumping speed, pump size) are selected until the rate requirement is fulfilled.

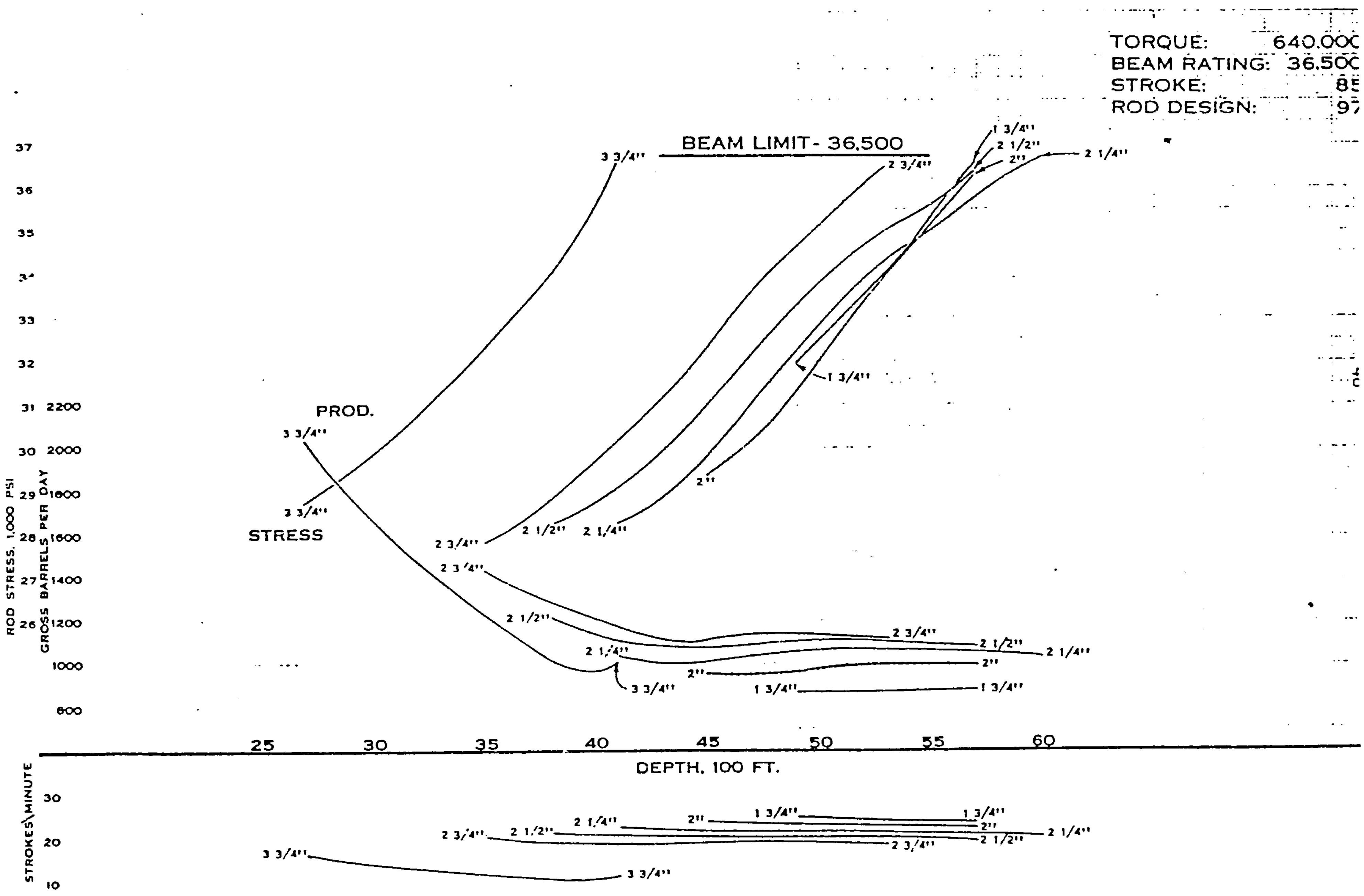
The loads are then calculated and used to select the appropriate pumping unit size, gear reducer and prime mover.

Selection charts and tables published in API BUL11L4 are very useful in determining initial choice of parameters.

COMPOSITE - 30,000 PSI STRESS LIMIT



	TORQUE:	STROKE:	ROD:
1	912,000	120	96
2	912,000	144	96
3	912,000	168	96
4	640,000	85	97
5	640,000	100	98
6	640,000	120	98
7	640,000	144	96
8	640,000	168	85
9	640,000	120	87
10	456,000	86	86
11	456,000	86	96
12	456,000	100	96
13	456,000	120	96
14	320,000	74	76
15	320,000	74	85
16	320,000	86	85
17	320,000	100	86
18	228,000	64	76
19	228,000	64	75
20	228,000	74	75
21	228,000	74	85
22	160,000	54	65
23	160,000	54	75
24	160,000	64	75
25	160,000	64	85
26	114,000	48	55
27	114,000	48	64
28	114,000	54	65
29	114,000	54	75
30	80,000	42	55
31	80,000	42	54
32	80,000	48	54
33	80,000	54	54
34	80,000	54	64
35	57,000	36	54
36	57,000	42	54
37	57,000	48	54



TORQUE: 640,00
BEAM RATING: 36,50
STROKE: 85
ROD DESIGN: 97
BEAM LIMIT- 36,500
PROD.
STRESS
3 3/4"
2 3/4"
2 1/2"
2 1/4"
2"
1 3/4"
ROD STRESS, 1,000 PSI
GROSS BARRELS PER DAY
37
36
35
33
32
31 2200
30 2000
29 1800
28 1600
27 1400
26 1200
1000
800
25
30
35
40
45
50
55
60
DEPTH, 100 FT.
STROKES\MINUTE
30
20
10

LUFKIN INDUSTRIES, INC.

F-989

PUMPING UNIT DESIGN CALCULATIONS

Company: ____________________ Well Name: ____________________ Date: __________

Field: ____________________ County: ____________________ State: __________

Req'd. Production: __________ BBL's./Day – Pump Depth __________ Ft. – Stroke Length __________ Inches

Plunger Dia.: __________ Inches – Tubing Size __________ Inches – Rod Size: __________ – Pumping Speed __________ SPM

ALL TYPES OF UNITS

1. Fo = Depth x Fluid Load, Table 1 = __________ x __________ = __________
2. SKR = 1000 x Stroke ÷ (Er, Table 2 x Depth) = 1000 x __________ ÷ (__________ x __________) = __________
3. Fo/SKR = __________ ÷ __________ = __________
4. N/No = SPM x Depth ÷ 245000 = __________ x __________ ÷ 245,000 = __________
5. N/No' = (N/No) ÷ Fe, Table 2 = __________ ÷ __________ = __________
6. BPD (100% eff.) = Pump Const. Table 1 x SPM x Stroke x SP, Table 3 = __________ x __________ x __________ x __________ = __________
7. WRF = Rod Weight, Table 2 · Depth = __________ x __________ = __________
8. WRF/SKR = __________ ÷ __________ = __________
9. TA = 1 + [%, Table 7 x ($\frac{WRF}{SKR}$ – .3) x 10] = 1 + [__________ x (__________ – .3) x 10] = __________

CONVENTIONAL UNITS

10. PPRL = WRF + (F1, Table 4 x SKR) = __________ + (__________ x __________) = __________
11. MPRL = WRF – (F2, Table 5 x SKR) = __________ – (__________ x __________) = __________
12. CBL = 1.06 x (WRF + Fo/2) = 1.06 x (__________ + __________) = __________
13. PT = T, Table 6 x SKR x Stroke/2 x TA = __________ x __________ x __________ x __________ = __________
14. Rod Stress = PPRL ÷ Area, Table 8 = __________ ÷ __________ = __________

AIR BALANCED UNITS

15. PPRL = WRF + Fo + .85 x (F1, Table 4 x SKR – Fo) = __________ + __________ + .85 x (__________ x __________ – __________) = __________
16. MPRL = PPRL – (F1, Table 4 + F2, Table 5) x SKR = __________ – (__________ + __________) x __________ = __________
17. CBL = 1.06 x (PPRL + MPRL) ÷ 2 = 1.06 x (__________ + __________) ÷ 2 = __________
18. PT = T, Table 6 x SKR x Stroke/2 x TA x .96 = __________ x __________ x __________ x __________ x .96 = __________
19. Rod Stress = PPRL ÷ Area, Table 8 = __________ ÷ __________ = __________

MARK II UNITS

20. PPRL = WRF + Fo + .75 x (F1, Table 4 x SKR – Fo) = __________ + __________ + .75 x (__________ x __________ – __________) = __________
21. MPRL = PPRL – (F1, Table 4 + F2, Table 5) x SKR = __________ – (__________ + __________) x __________ = __________
22. CBL = 1.04 x (PPRL + 1.25 x MPRL) ÷ 2 = 1.04 x (__________ + 1.25 x __________) ÷ 2 = __________
23. PT = (PPRL x .93 – MPRL x 1.2) x Stroke ÷ 4 = (__________ x .93 – __________ x 1.2) x __________ ÷ 4 = __________
24. Rod Stress = PPRL ÷ Area, Table 8 = __________ ÷ __________ = __________
25. NOTE: Do Not Use Less Than One Size Smaller Reduced Than Required For Conventional Unit

March 1973

116

Pumping System Performance

The principal objectives for determining performance are:

a) Determine whether the system is operating according to design

c) Identify changes in operating conditions that require operator's attention.

d) Identify causes of performance change.

The two main techniques for performance analysis are the Dynamometer Survey and the Fluid Level Survey.

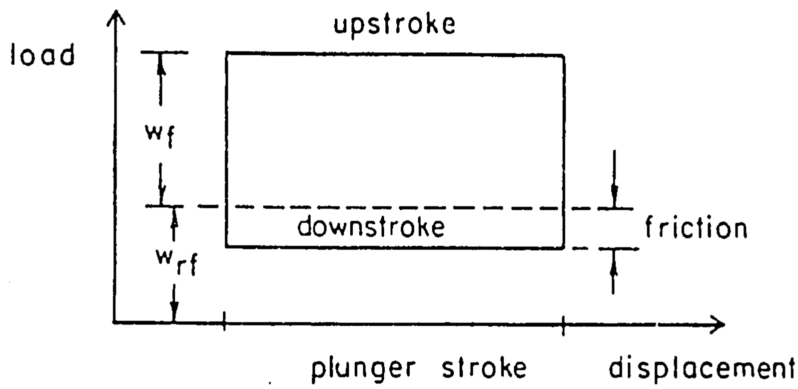
Dynamometer Survey

The load-displacement curve is measured at the polished rod during one or more complete pumping cycles. The resulting diagram is compared with theoretical or with previously measured diagrams in order to identify abnormal characteristics.

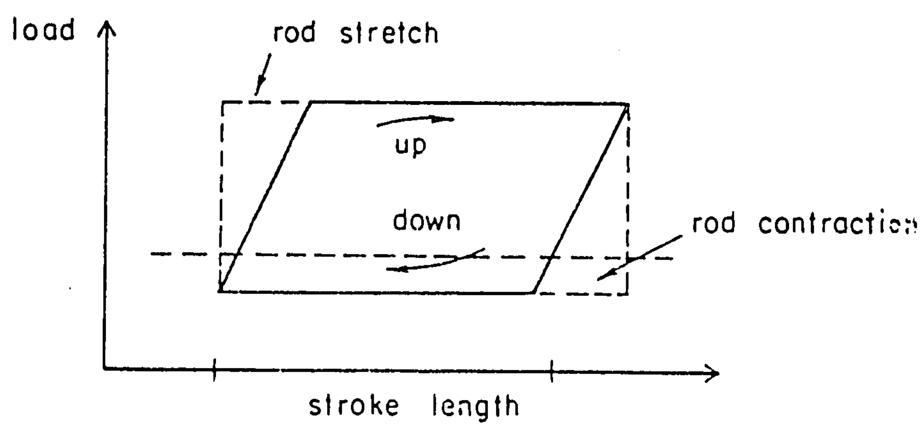
Ideal Dynamometer Diagram

The following schematic diagrams illustrate the various components that combine in the dynamometer diagram.

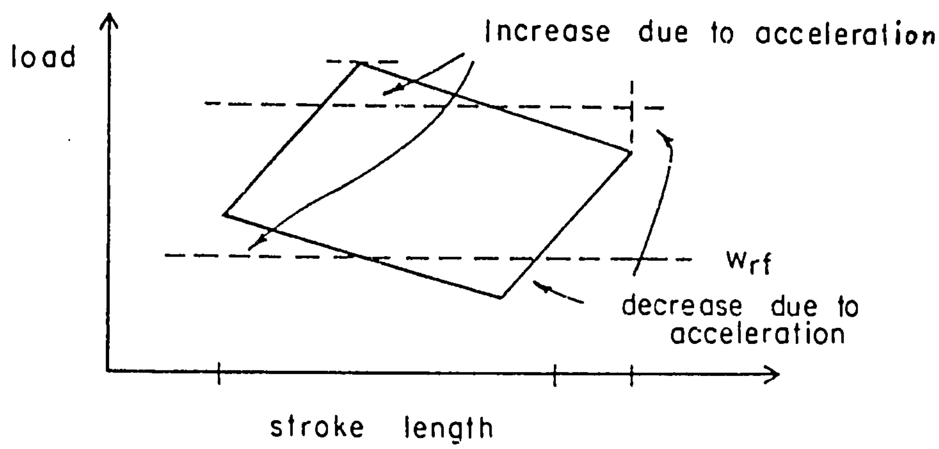
Load-displacement curve, measured at the polished rod for inelastic, "zero" speed case.



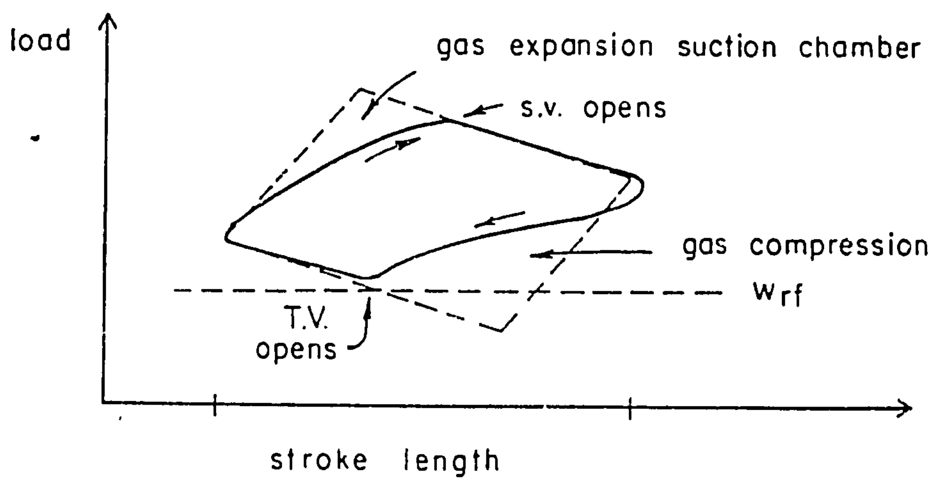
a) Influence of rod stretch and contraction



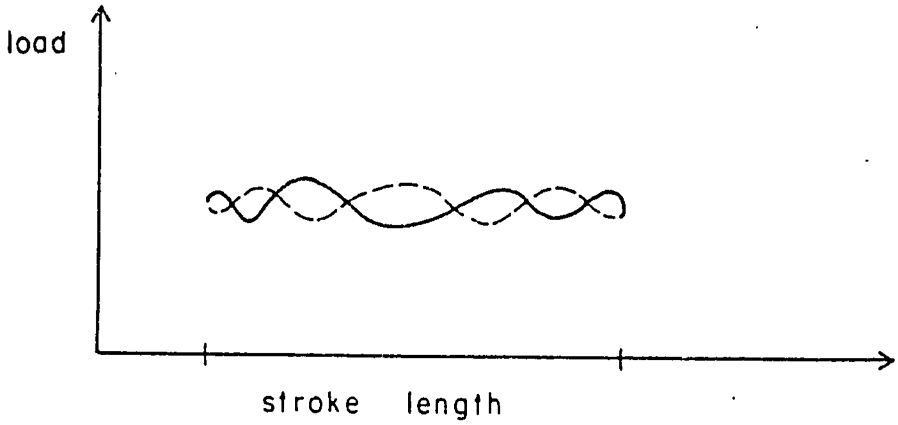
b) Influence of acceleration



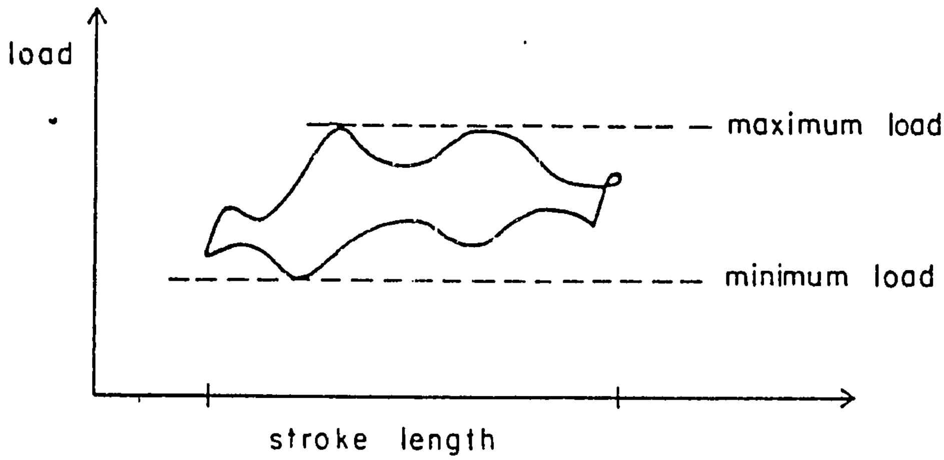
c) Influence of gas compression and expansion



d) Influence of rod vibrations



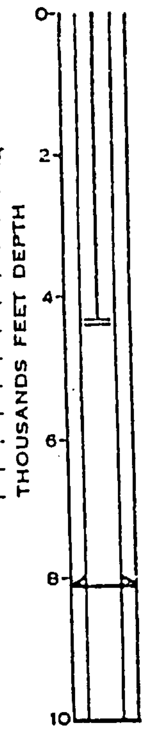
which when superposed to curve C gives



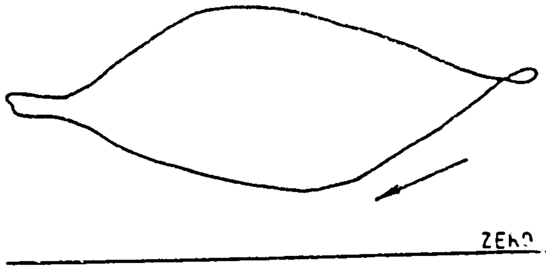
is is the diagram measured at the polished rod.

Atlantic Field - Arkansas

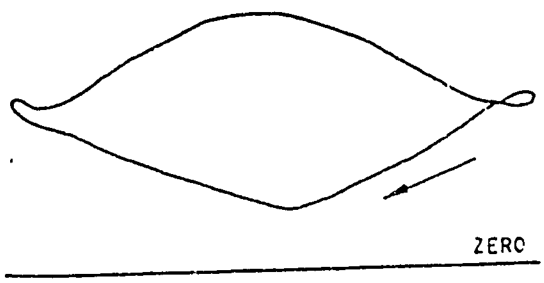
TOTAL DEPTH 8188 FT FORMATION SHACKOVER LIME
DAILY PRODUCTION: OIL 13 WATER 270 SP. GRAV. 44°
STROKE LENGTH 64 IN. S.P.M. 18 A.V.E. 68.9 PCT
FLUID LEVEL: STATIC ____ PUMPING ____ B.H.P. ____
WELL CLASSIFICATION: AGITATOR ____ PUMPER X
DOES WELL POUND No CASING HEAD PRES. 50 LB
CORROSIVE CONDITIONS: PITTING YES H2S YES
CASING SIZE 5½ IN. FEET 8186 TUBING SIZE 2 IN.
GAS ANCHOR ____ TUBING ANCHORED No
SUCKER RODS: 1" ____ 7/8" ____ 3/4" 4275 FT 5/8" ____
TYPE STEEL C .07-.11; MAN .20-.50; SI .15-.30; NI 3.00-3.50; CU 1.45-1.65
PUMP 2 IN. X 1-3/4 IN. X 9 FT TW PLGR. SIZE 1-3/4 IN.
PUMPING UNIT TWIN CRANK MOTOR 25 HP ELECTRIC
UNIT RATING: LOAD 25,000 LB PEAK TORQUE 285,620 IN.-LB
GEAR BOX: SINGLE ____ DOUBLE X RATIO ____



THESE DYNAMOMETER CARDS SHOW VERY SMOOTH OPERATION. THIS MEANS THE SUCKER RODS HAVE A REASONABLE CHANCE FOR TROUBLE-FREE OPERATION IN SPITE OF THE CORROSIVE FLUID CONDITIONS. UNIT STRESS IS 26,000 PSI.



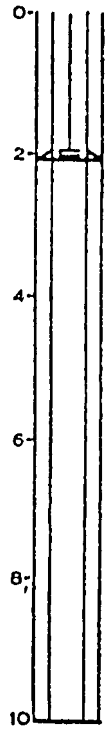
MAX LOAD	11,500 LB
MIN LOAD	3,800 LB
RANGE	7,700 LB
SPEED	18 SPM
STROKE	64 IN.
POL ROD HP	11.02
ENGINE RPM	1170
TIME	11:00 AM



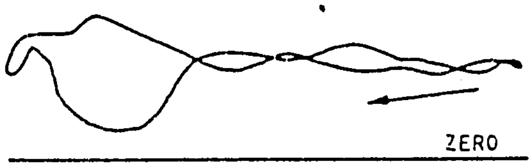
MAX LOAD	11,500 LB
MIN LOAD	3,800 LB
RANGE	7,700 LB
SPEED	18 SPM
STROKE	64 IN.
POL ROD HP	10.64
ENGINE RPM	1170
TIME	11:15 AM

Boyd Field - Illinois

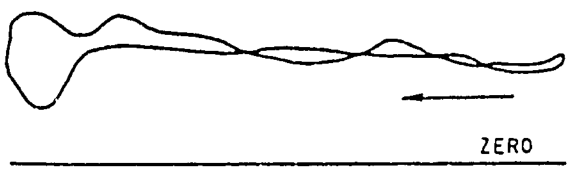
TOTAL DEPTH 2044 FT FORMATION BENOIST SAND
DAILY PRODUCTION: OIL 24 WATER 43 SP. GRAV. 39°
STROKE LENGTH 42 IN. S.P.M. 20 A.V.E. 22.3 PCT
FLUID LEVEL: STATIC ____ PUMPING LOW B.H.P. ____
WELL CLASSIFICATION: AGITATOR ____ PUMPER X
DOES WELL POUND YES CASING HEAD PRES. NONE
CORROSIVE CONDITIONS: PITTING NO H2S NO
CASING SIZE 5½ IN. FEET 2025 TUBING SIZE 2 IN.
GAS ANCHOR NO TUBING ANCHORED NO
SUCKER RODS: 1" ____ 7/8" ____ 3/4" 2024 FT 5/8" ____
TYPE STEEL 1035 CARBON
PUMP 2 IN. x 1-3/4 IN. x 7 FT TW PLGR. SIZE 1-3/4 IN.
PUMPING UNIT TWIN CRANK MOTOR 16 HP - 4-CYL GAS
UNIT RATING: LOAD 10,450 LB PEAK TORQUE 57,000 IN.-LB
GEAR BOX: SINGLE ____ DOUBLE X RATIO 29.33/1



THESE CARDS ARE CHARACTERISTIC OF LOW PRODUCTION OR STRIPPER WELLS. IN THIS TYPE OF WELL IT IS OFTEN DIFFICULT TO DECREASE THE PUMP DISPLACEMENT ENOUGH TO COMPLETELY ELIMINATE THE FLUID POUND. A 1½ IN. PUMP OPERATING AT 20 TO 24-42 IN. SPM SHOULD PRODUCE THE MAXIMUM VOLUME THIS WELL WILL MAKE.

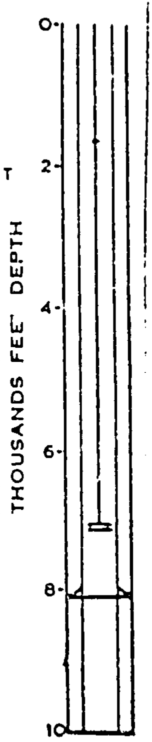


MAX LOAD	6,800 LB
MIN LOAD	1,800 LB
RANGE	5,000 LB
SPEED	20 SPM
STROKE	42 IN.
POL ROD HP	1.71
TIME	11:30 AM



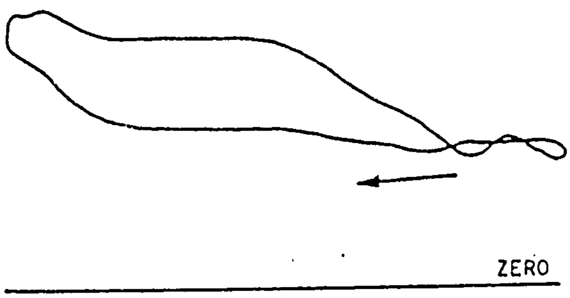
MAX LOAD	7,200 LB
MIN LOAD	3,000 LB
RANGE	4,200 LB
SPEED	20 SPM
STROKE	42 IN.
POL ROD HP	1.14
TIME	12:15 PM

Livingston Field - South Texas

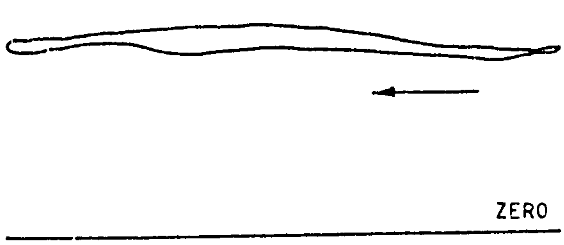


TOTAL DEPTH 8150 FT FORMATION WILCOX
DAILY PRODUCTION: OIL 50 WATER 5 SP. GRAV. 37°
STROKE LENGTH 64 IN. S.P.M. 12 A.V.E. 39.2 PCT
FLUID LEVEL: STATIC 5086 FT PUMPING ______ B.H.P. ______
WELL CLASSIFICATION: AGITATOR X PUMPER ______
DOES WELL POUND NO CASING HEAD PRES. 30 LB
CORROSIVE CONDITIONS: PITTING NO H2S NO
CASING SIZE $5\frac{1}{2}$ IN. FEET 7125 TUBING SIZE 2 IN.
GAS ANCHOR 10 FT $1\frac{1}{2}$ IN. TUBING ANCHORED NO
SUCKER RODS: 1" ______ 7/8" ______ 3/4" 1650 FT 5/8" 5475 FT
TYPE STEEL 4620 NICKEL MOLYBDENUM
PUMP 2 IN. X $1\frac{1}{2}$ IN. X 15 FT RWA PLGR. SIZE $1\frac{1}{2}$ IN.
PUMPING UNIT TWIN CRANK MOTOR MULTI-CYL GAS
UNIT RATING: LOAD 20,000 LB PEAK TORQUE 250,000 IN.-LB
GEAR BOX: SINGLE ______ DOUBLE X RATIO ______

THIS WELL IS AGITATING AND FLOWING AND FOR THAT REASON SHOULD BE PUMPED AT A SLOW SPEED. HIGHER SPEEDS OF OPERATION WILL ONLY PUT INCREASED LOADS ON THE SUCKER ROD STRING AND SURFACE EQUIPMENT.



MAX LOAD	13,000 LB
MIN LOAD	6,600 LB
RANGE	6,400 LB
SPEED	17 SPM
STROKE	64 IN.
POL ROD HP	6.1
TIME	8:05 AM



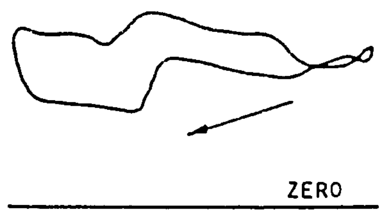
MAX LOAD	10,200 LB
MIN LOAD	9,000 LB
RANGE	1,200 LB
SPEED	13 SPM
STROKE	64 IN.
POL ROD HP	0.9
TIME	8:35 AM

Gaines County Field - West Texas

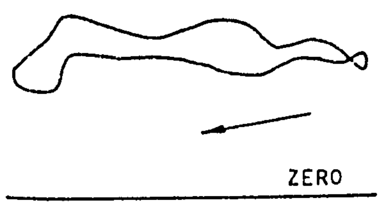
TOTAL DEPTH 5300 FT FORMATION LIME
DAILY PRODUCTION: OIL 10* WATER 6 SP. GRAV. ____
STROKE LENGTH 44 IN. S.P.M. 16 A.V.E. 23.1 PCT
FLUID LEVEL: STATIC ____ PUMPING LOW B.H.P. ____
WELL CLASSIFICATION: AGITATOR ____ PUMPER X
DOES WELL POUND YES CASING HEAD PRES. 10 LB
CORROSIVE CONDITIONS: PITTING NO H2S YES
CASING SIZE $5\frac{1}{2}$ IN. FEET ____ TUBING SIZE 2 IN.
GAS ANCHOR 6 FT $1\frac{1}{2}$ IN. TUBING ANCHORED NO
SUCKER RODS: 1" ____ 7/8" ____ 3/4" 5175 FT 5/8" ____
TYPE STEEL 4620 NICKEL MOLYBDENUM
PUMP 2 IN. X $1\frac{1}{2}$ IN. X 12 FT RWB PLGR. SIZE $1\frac{1}{2}$ IN.
PUMPING UNIT TWIN CRANK MOTOR 25 HP MULTI-CYL
UNIT RATING: LOAD 17,000 LB PEAK TORQUE 144,520 IN.-LB
GEAR BOX: SINGLE ____ DOUBLE X RATIO ____

*OPERATES 9 HOURS PER DAY.

THIS WELL IS A PART-TIME PUMPER AND ILLUSTRATES THE PROGRESSIVE PUMP-OFF OF A WELL. A TAPERED STRING OF 3/4-INCH AND 5/8-INCH RODS SHOULD WORK SATISFACTORILY IN THIS CASE, THEREBY DECREASING THE UNIT STRESS ON THE TOP SECTION OF THE 3/4-INCH RODS.

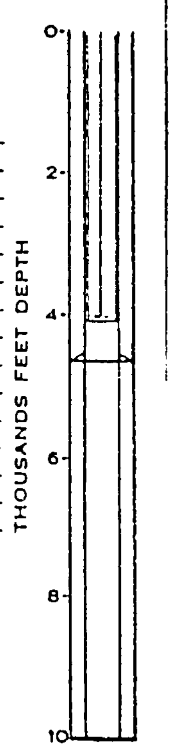


MAX LOAD	13,850 LB
MIN LOAD	6,800 LB
RANGE	7,050 LB
SPEED	16 SPM
STROKE	44 IN.
POL ROD HP	5.5
TIME	9:00 AM



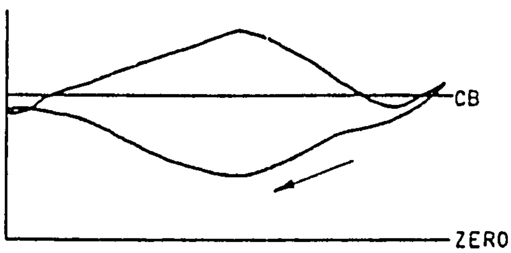
MAX LOAD	13,300 LB
MIN LOAD	7,100 LB
RANGE	6,200 LB
SPEED	19 SPM
STROKE	44 IN.
POL ROD HP	4.6
TIME	11:30 AM

South Mankins Field - North Texas

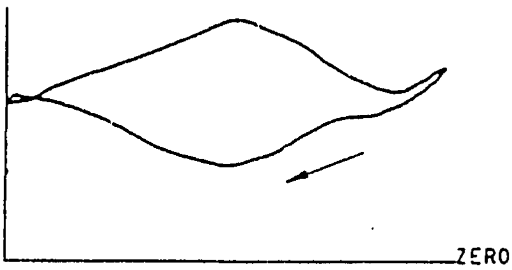


TOTAL DEPTH 4684 FT FORMATION CADDO LIME
DAILY PRODUCTION: OIL 14 WATER 200 SP. GRAV. ____
STROKE LENGTH 56 S.P.M. 16 A.V.E. 67 PCT
FLUID LEVEL: STATIC ____ PUMPING ____ B.H.P. ____
WELL CLASSIFICATION: AGITATOR ____ PUMPER X
DOES WELL POUND No CASING HEAD PRES. 20 PSI
CORROSIVE CONDITIONS: PITTING MILD H2S No
CASING SIZE 7 IN. FEET 4656 TUBING SIZE 2½ IN. EUE
GAS ANCHOR 1 IN. x 10.5 FT TUBING ANCHORED No
SUCKER RODS: 1" ____ 7/8" ____ 3/4" 4056 FT 5/8" ____
TYPE STEEL CARBON STEEL
PUMP 2½ IN. x 1-3/4 IN. x 14 FT PLGR. SIZE 1-3/4 IN.
PUMPING UNIT TWIN CRANK MOTOR GAS ENGINE 25 HP
UNIT RATING: LOAD 15,000 LB PEAK TORQUE 113,000 IN.-LB
GEAR BOX: SINGLE ____ DOUBLE X RATIO 30.6/1

THE GENERAL SHAPE OF THESE CARDS IS VERY CLOSE TO THE THIRD ORDER HARMONIC. THE TRAVELING VALVE IS OPENING AT THE BEGINNING OF THE DOWNSTROKE, AND THE FULL-BODIED CARDS ARE CONSISTENT WITH WELLS MAKING A HIGH PERCENTAGE OF WATER. IT SHOULD BE POSSIBLE TO INCREASE THE APPARENT VOLUMETRIC EFFICIENCY BY USING A 7/8 IN.-3/4 IN. TAPERED SUCKER ROD STRING.



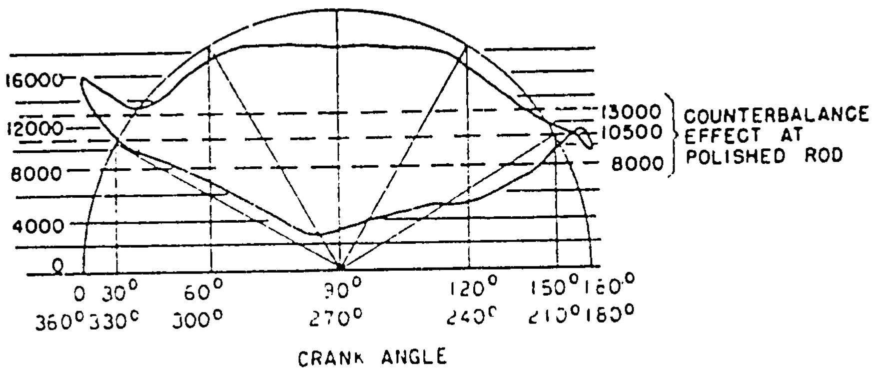
MAX LOAD	13,200 LB
MIN LOAD	3,800 LB
RANGE	9,400 LB
SPEED	16 SPM
STROKE	56 IN.
POL ROD HP	9.3



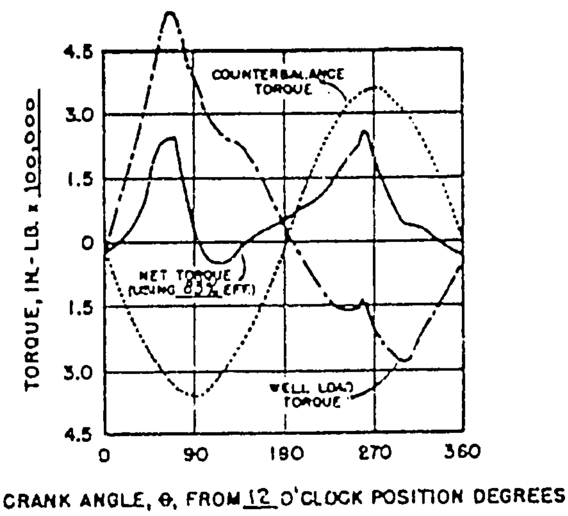
MAX LOAD	13,200 LB
MIN LOAD	3,800 LB
RANGE	9,400 LB
SPEED	16 SPM
STROKE	56 IN.
POL ROD HP	9.3

Dynamometer curve can be analyzed qualitatively to indicate possible malfunctions of the system. (See atached samples)

It can also be analyzed analytically to determine torque loading as a function of crank angle.



Torque factor depends on unit geometry. Torque is positive on upstroke and negative on downstroke.



Computer programs are available to interpret dynamometer information and by removing the dynamic effects of the rod string obtain an equivalent "dynagraph" representing the performance of the downhole pump.

Continuous Monitoring of Well Load

In automated production facilities (CPC Systems) it is possible to monitor continually the well load.

Use beam load monitor (strain gauges) or polished rod load cell.

Usually position information is not monitored, but some systems also include an angular position transducer that monitors angular displacement of beam.

Load monitoring allows determination of well pump-off -- determination when unit can be shut down to save power. Also gives indication of malfunctions:

a) Rod failures
b) Pump wear
c) Friction
d) Pump efficiency

The same system is available in portable form allowing fast diagnosis of well which are not connected to CPC Systems.

Fluid Level Survey

Performance analysis of pumping wells requires determination of annulus fluid level. This in turn allows determination of flowing bottom-hole pressure. In some key wells, permanent bottom-hole pressure recorders are installed, that give surface indications of bottom-hole pressure.

The problem of determining fluid level has been solved by means of acoustic surveys (Sonolog). Since pressure affects sound velocity the determination relies on tubing collar reflections. Curves are also available that include effect of pressure and temperature on gas acoustic velocity.

The following diagram represents a typical record obtained from an Echometer Survey. (Next page).

Flowing BHP Calculations

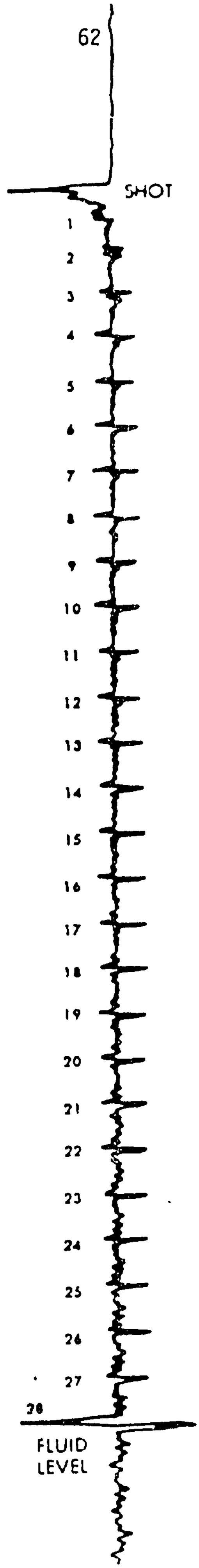
The conversion of fluid level to bottom-hole pressure is complicated by the difficulty in establishing the gradient of the fluid in the annulus.

Method 1

Assume that fluid gradient is not affected by pressure.

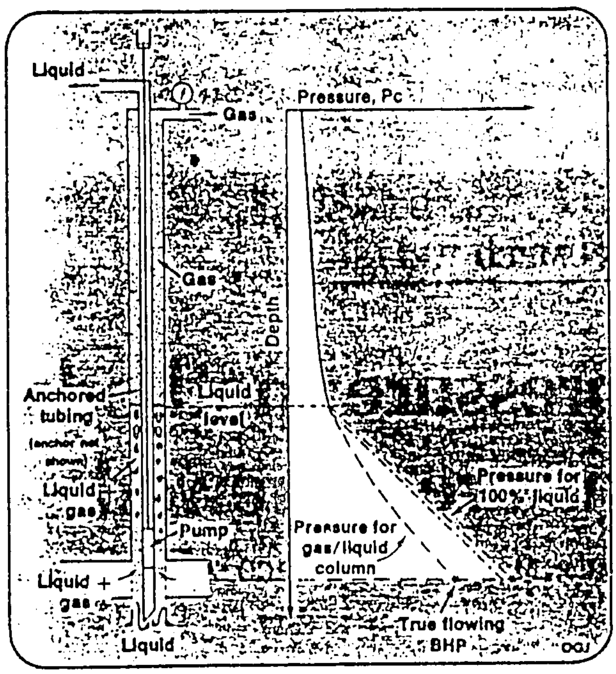
Two fluid levels are taken with different back pressure in the annulus.

Measurements are taken sufficiently far apart to insure that steady state conditions of pumping are achieved.

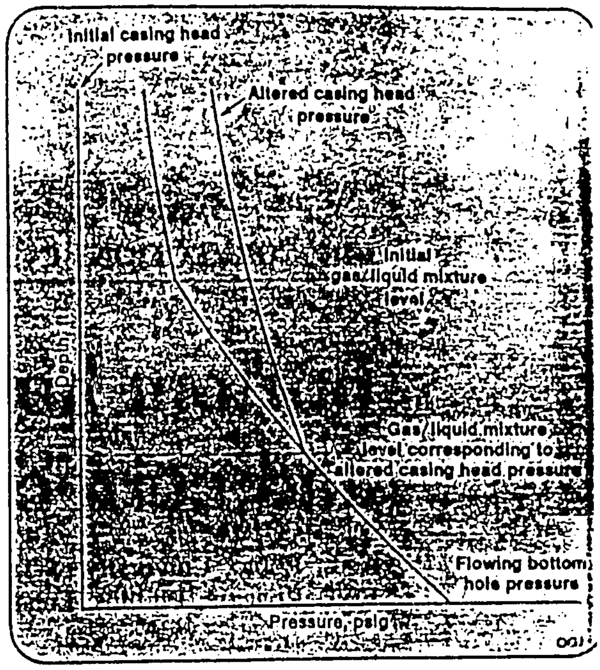

62
SHOT
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
19
20
21
22
23
24
25
26
27
28
FLUID
LEVEL

JBJECT

Typical completion & pressure traverse



Walker's method



It is imperative that the same fluid production be obtained at the two casing back pressures.

Method 2

Close casing to depress fluid level to the pump, until well pumps-off. Then

$$P_{wf} = CHP + Gg\ D$$

The production rate is disturbed by the flow of gas into the tubing.

where:

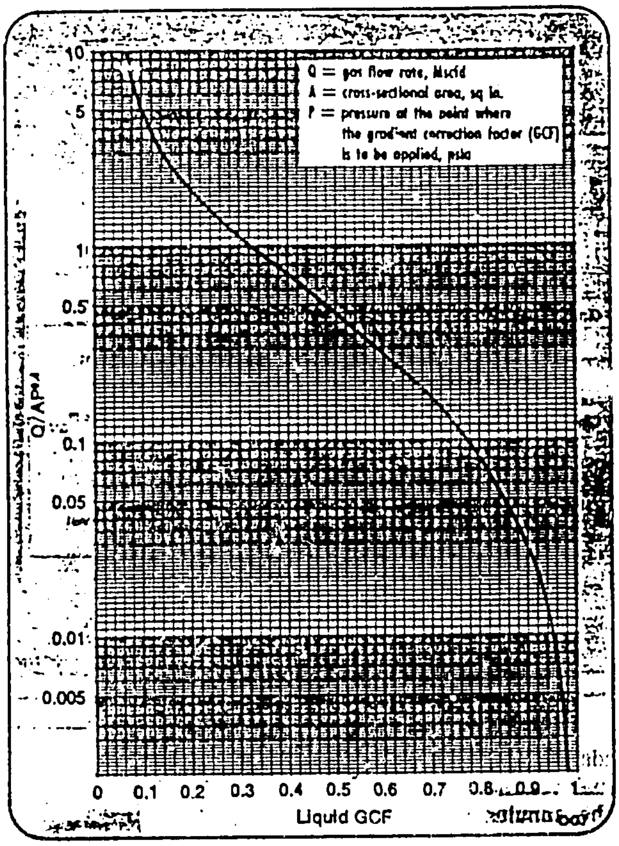
CHP + Casinghead pressure (psi)

Gg = Gas gradient (psi)

Method 3

Calculation of the casing liquid gradient based on tl free liquid and the valume of gas that is being annulus. The ratio of mixture density to liquid density has been correlated with $\frac{q}{AP^{0.4}}$.

"S" curve for liquid GCF



R_m

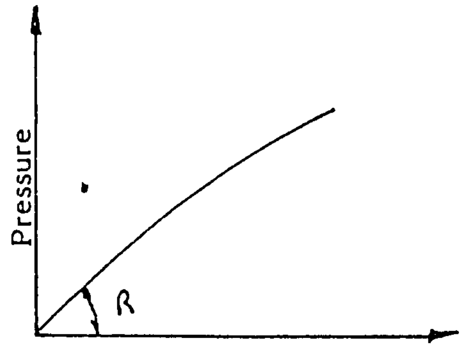
$$R_m = \frac{\text{mixture density}}{\text{liquid density}}$$

at pressure P (psia)

This allows determining the gradient of the mixture. Since the pressure is not known at a given point, successive approximations are required corresponding to fixed pressure increments.

The flow of gas q, can be measured with a critical flow prover, or it can be determined from a pressure build-up of the casing which is shut in after taking the sonolog.

131



$$R = \frac{\text{Pressure}}{\text{time}} \text{ (psi/min)}$$

$$\frac{q}{A} = 0.68 \quad R\left(\frac{D}{1000} + \frac{P_g}{433 \; G} \right) \quad \frac{\text{mcf}}{\text{day-sqin.}}$$

D = Depth to liquid feet

Pg = gas pressure at D

G = gas gravity

For low pressure

$$Pg = (.1 + \frac{D}{40000}) P_c \quad \text{psia}$$

Taking a pressure interval $P \approx 50$ 100 psi calculate P_{av} and $\frac{q}{A \; P_{av}^{0.4}}$ to get R_m from where

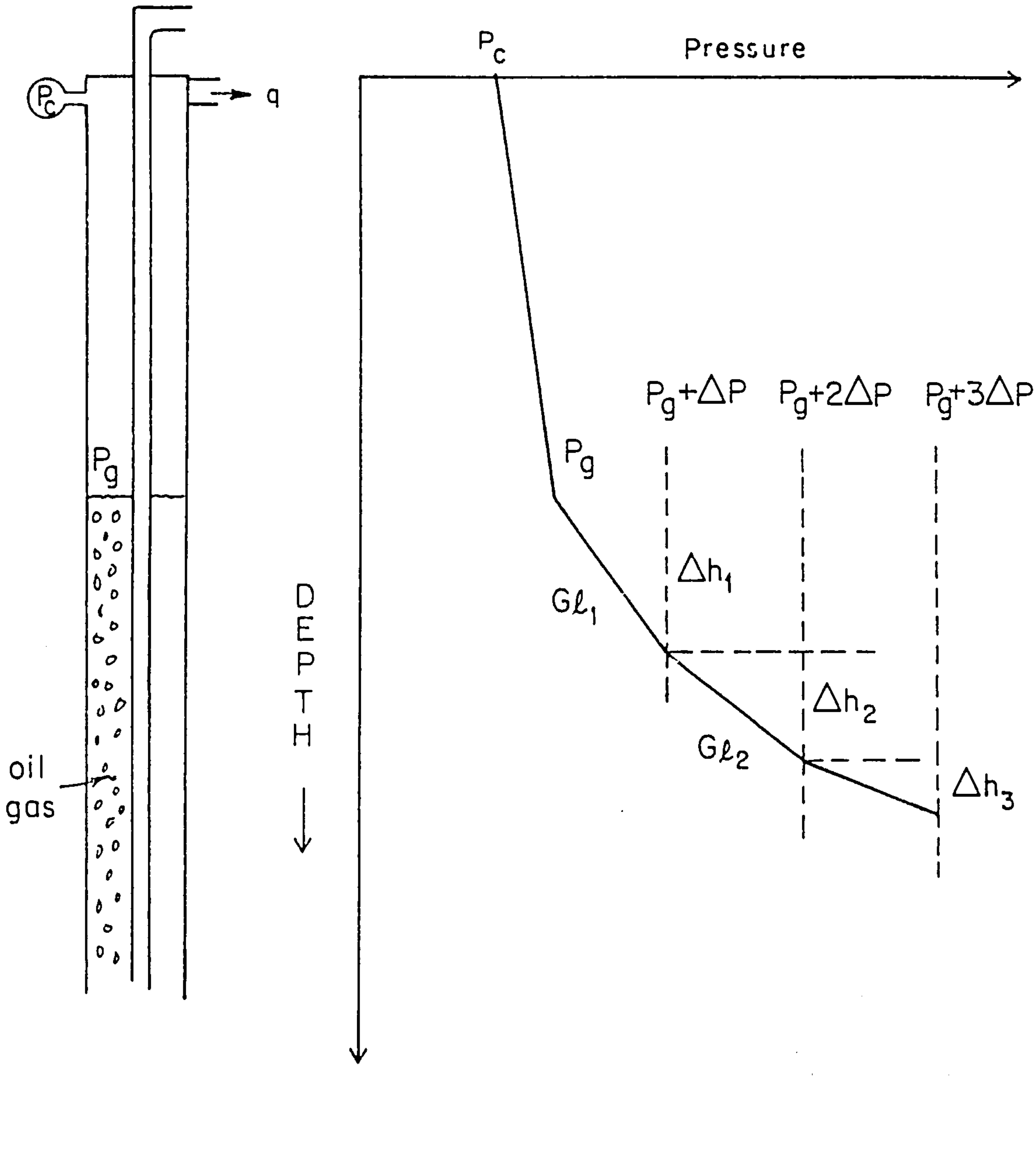
$$G_\ell = 0.433 \times R_m \times \gamma_\ell \quad \text{psi/ft}$$

γ_ℓ = specific gravity of gas-free liquid.

$$\Delta_h = \frac{P_2 - P_1}{G}$$

132

The following diagram illustrates the step-wise calculation:



139

SUBMERSIBLE PUMPS

INTRODUCTION

An electric submersible puming system is generally considered to be a high volume type of artificial lift. It is most applicable in wells that are under the influence of a water drive or waterflood and that have high water cuts or low GOR's. This application is based on two key characteristics of the submersible pumping system:

(1) The system can efficiently deliver the largest amount of horsepower at the pump of any pumping system in the small diameter casing sizes used in oilfield applications.

(2) Centrifugal pumps can produce at much higher rates than positive displacement pumps in wells of limited diameters.

The submersible pump is a multistage electrical centrifugal pump that operates completely submerged in the fluid it pumps. The major components of the system are the pump, the motor, the seal section or protector, cable, and surface equipment such as transformer and switchboard. A typical submersible installation is shown in Figure 1.

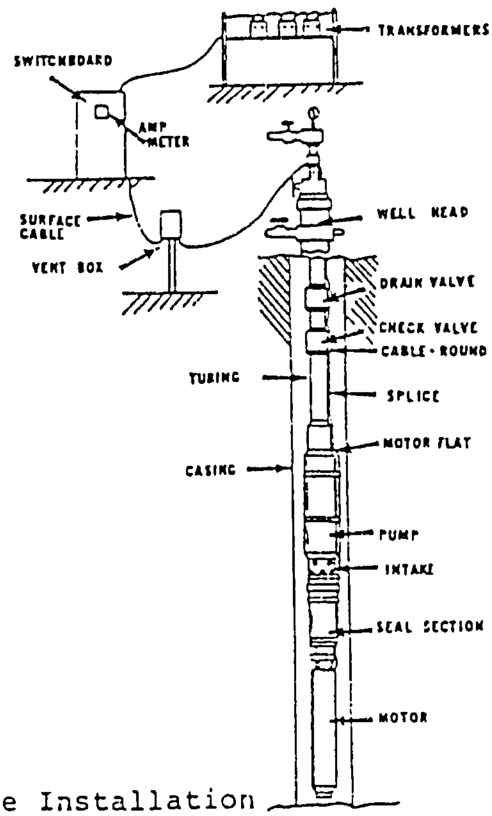


Figure 1.
Typical Submersible Installation

Over recent years the industry has gained considerable experience in producing high viscosity fluids, gassy wells and high temperature wells using submersible pumps. With this experience and improved technology, wells that were once considered difficult for submersibles are now being pumped economically.

Submersible systems are capable of lifting from 200 B/D up to 60,000 B/D at depths of 1000 ft to 15,000 ft. They are suitable for both onshore and offshore wells in deviated or straight holes.

COMPONENTS

Motor

The submersible electric motor is a 3-phase, induction type, that is oil-filled for cooling and lubrication. A high starting torque enables it to reach full load operating speed of 3475 rpm in less than fifteen cycles, thus reducing drag on the power supply. Cooling is achieved by heat transfer to well fluid moving by the motor. For this reason, the unit is usually set above the producing interval. If the motor is to be set within or below the producing interval, a cooling jacket (shroud) over the motor and seal section is recommended to force the produced fluid to travel down and then inside the shroud. In this manner the necessary cooling can be maintained.

Seal Section or Protector

The seal section performs four basic functions which are:

1. Connects the pump housing to the motor housing by connecting the drive shaft of the motor to the pump shaft.
2. Seals the power end of motor from the wellbore fluids while allowing pressure communication between the motor and wellbore fluids.
3. Houses a marine-type thrust bearing which absorbs axial loading from the pump.
4. Provides the necessary volume for expansion of the motor's oil which results from heat generated when the motor is in operation.

Gas Separator or Intake Section

The gas separator is a bolt-on section between the protector and the pump where it serves as the pump intake. It is designed to separate a substantial portion of any free gas in the produced fluid. Through the action of a gas-lock free, open faced impeller, it provides optimum charging of the pump. For water or applications with only small amounts of free gas, a bolt-on direct intake section allowing direct entry to the first stage can be provided. Production of high gas-oil-ratio reservoirs in discussed in a later section.

Pump

The multistaged centrifugal pump is offered in a wide variety of diameters. Each stage consists of a rotating impeller and stationary diffuser. Through the use of corrosion resistant materials, (cast Ni-resist or molded nonmetallic polyphenylene sulfide impellers and diffusers with K-monel shafting) pump wear and corrosion are minimized. Because of the hydraulic limitations associated with casing diameter restrictions, the lift per stage is relatively low. However, as many as 500 or more stages have been run to meet high head requirements.

Electric Cable

Power is transmitted by an electric cable specially designed and built to strict specifications for submersible applications. A range of conductor sizes permits efficient matching to motor requirements and can be obtained in either a flat or round configuration. Round cable is normally recommended; however, flat cable can be used where clearance is a problem.

The insulation used for these cables must be able to withstand well temperatures, chemical attack, pressure, and resist impregnation of well fluids. Cables may now be installed in wells with temperatures in excess of 275°F. Mechanical protection is provided by an interlocking armor of steel or monel, as required by well conditions.

Junction Box

As shown in Figure 1, the weatherproof, vented junction box provides a convenient and safe means of connecting the downhole cable to a surface cable from the switchboard. The junction box also provides a vent to the atmosphere for any gas that might migrate up the power cable.

Switchboard

The switchboard controls the operation of the motor. The standard switchboards are weatherproof and available in a range of sizes and accompanying accessories to accommodate any installation.

Simple units may contain only push-button, magnetic contactors and overload protection. More sophisticated switchboards will use solid state motor controllers for time-delayed underload protection on all three phases, time delayed overload protection, and automatic protection against voltage unbalance conditions. Most switchboards will also contain recording ammeters to give immediate indications of problems or abnormal operation.

Downhole Pressure Monitors

Another valuable device to include is a "donhole-pressure monitor" strip chart recorder. Valuable reservoir and pump performance data is available with the use of downhole-pressure monitors. By correlating reservoir pressure with the withdrawal rate, an operator can determine the need to change pump size, change injection rate, or consider a well workover.

CENTRILIFT's "PHD system" has the capability of:

1. Continuously monitoring bottom hole pressure at the pump's setting depth.

2. Protecting the motor against overheating.

3. Detecting electrical failures such as shorts to ground.

The system requires no special wires. All signals are sent to the surface instruments over the regular power cable.

Check Valve

A check valve is usually installed in the tubing string two or three joints above the pump assembly. This prevents the tubing from unloading through the pump when the unit goes down. An attempt to start the unit with the pump rotating backward could break the pump shaft, burn out the motor, or burn out the cable. Many check valves can be removed by wireline for service, or to pump treating fluids down the tubing and through the pump.

Drain Valve or Sliding Sleeve

A sliding sleeve is installed one joint above the check valve for two main purposes:

1. It allows communication between tubing and annulus when required for injecting chemical treatments into the formation.

2. It allows the tubing to be pulled dry if the pump becomes plugged or if a check valve is in the string.

Wellhead

The wellhead must be equipped with a tubing head bonnet or pack-off which provides for a positive seal around the cable and the tubing. There are several methods available from well manufacturers for providing this pack-off. Depending on method used, the pack-off will be capable of holding from 500 psi pressure in the simple pack-off systems to over 3000 psi in the more sophisticated systems.

Transformer

Transormer selection depends on the primary power system and the required surface voltage and current. Banks of three single-phase transformers or three-phase auto transformers are available. If the well might require a larger pump unit in the future, it is most economical to install the larger size transformer initially.

DESIGN PROCEDURE

Sizing of submersible pumping equipment is critical. Each model of pump has a limited throughput range within which it will operate without creating excessive wear due to thrust between the pump diffusers and floating impellers.

The design of a submersible pumping unit, under most conditions, is not a difficult task if reliable data is available. But if the data, especially that pertaining to the well's capacity, is poor, the design will usually be marginal. In such cases, a pump is usually sized incorrectly. An oversized pump will run in downthrust and result in excessive thrust bearing and stage wear, thus reducing the life of the equipment. Units operating in a pumped-off condition can accelerate cable failure, especially the flat cable. Here, without liquid to help dissipate the heat, the temperature can become excessive.

High Water Cut Wells

In sizing a submersible unit for a high WOR application the following seven-step procedure should be used:

1. Collect and analyze well production, fluid, and electrical power data.

2. Determine the well productivity at the desired pump setting depth <u>or</u> determine the pump setting depth for the desired production rate.

3. Calculate the "Total Dynamic Head" (TDH).

4. For given capacity, select the pump type which will have the highest efficiency for the volume.

5. From manufacturer's pump performance curves and motor data sheet, determine the optimum size of the pump, motor, and seal section.

6. Select electric cable size and type.

7. Select motor controller, transformer and accessories.

Collecting Data

The first step is to obtain well productivity data. Accurate determination of the well's capacity to produce at the desired pumping depth is the major factor in the proper selection of submersible pumping systems. Data needed include producing rate, static bottom-hole well productivity index (PI), inflow performance relationship (IPR), bottom-hole temperature, well-head pressure, gas-oil ratio, water-oil ratio, API gravity and specific gravity of the produced fluid, bubble point pressure of the gas, oil viscosity and any other special operating conditions such as sand, corrosion, paraffin, or emulsion problems.

As discussed under a previous section dealing with "Well Performance", most wells will not exhibit a true straight line Productivity Index relationship due to gas interference and turbulence in the bore area. For this reason it is generally recommended that the Vogel technique for Inflow Performance Relationship be used to determine productivity. The importance of accurately defining a well's productivity cannot be over-emphasized.

Total Dynamic Head

The next step is to determine the total head required to pump the desired capacity. This refers to the feet of liquid being pumped and is the sum of:

1. Net feet of fluid lift (h_d) - distance from the wellhead to the estimated producing fluid level at expected producing capacity.

2. Friction loss in the tubing (f_t).

3. Tubing discharge pressure (p_d) - the head needed to overcome friction in the surface pipe, valves, and fittings and to overcome elevation changes between wellhead and tank battery.

The simplified equation for Total Dynamic Head is:

$$H = h_d + f_t + p_d$$

Pump Selection

Choice of the pump is based on estimated fluid producing rates and casing size. The largest diameter pump which the casing will permit is usually the most economical choice. The pump selected should have the desired capacity within its optimum limits and nearest its peak efficiency.

After selecting the correct pump, the number of stages required to produce the anticipated cpacity against the previously calculated total dynamic head can be made using the pump performance curve. This may be calculated as follows by reading the head per stage and the desired pumping rate:

$$\text{Number of Stages} = \frac{\text{Total Dynamic Head (ft)}}{\text{Head per Stage (ft/stage)}}$$

Motor Selection

Submersible pump motors are available in a wide range of horsepowers and operating voltages. The required motor size for a predetermined pump size is proportional to the number of stages and the specific gravity of the produced fluid. The equation may be written:

HP or BHP = No. of Stages x HP/Stage x Sp. Gr.

The motor voltage selected for a given installation is dependent on available surface voltage, casing-tubing dimensions, economics, and future horsepower requirements. The required surface voltage is the sum of the cable IR drop and the rated motor operating voltage. When the voltage loss becomes too great, a higher voltage (lower amperage) motor is required.

Generally, when the pump setting depth is 5,000 ft or more, a higher voltage motor will yield the best economics for an installation. The decrease in cable size and cost resulting from the lower current requirements of the higher voltage motor will offset the cost of the more expensive switchboard.

Cable Size and Length

The proper cable size is determined by combined factors of voltage drop, amperage and available space between tubing collars and casing. Electric cables are availale in conductor sizes from No. 1 through No. 8 with copper conductors. Usual practice limits the voltage drop to less than 30 volts/1000 ft.

The total cable length should be at least 100 ft longer than the pump setting depth in order to make surface connections a safe distance from the wellhead.

Switchboard

Switchboards are offered in a range of voltages from 440 through 5000 volts. Selection is based upon ratings of voltage, amperage, horsepower and future requirements.

Transformers

The type of transformer required depends on the primary power system and the required surface voltage. Three-phase auto-transformers are generally required for increasing voltages fom a 440/480 volt system to an 800 to 1000 volt range. A bank of three single-phase transformers is usually needed for reducing the higher voltage primary to the required surface voltage if they are below the 440/480 volt level.

In choosing either type of transformer, the following equation is recommended:

$$KVA = \frac{1.73 \times V_s \times A_m}{1{,}000}$$

Where: KVA = 1000 volt-amperes

V_s = Surface Voltage ($V_m + V_l$)
This value is the sum of motor nameplate voltage (V_m) and the voltage loss (V_l) in the electric cable between the motor and the switchboard

A_m = Motor nameplate current in Amps

If the well might require a larger pumping unit in the future, it is more economical to install a larger size transformer bank initially.

EXAMPLE PROBLEM

For a detailed illustration of the correct procedure to follow in sizing a submersible centrifugal pumping unit for a high water cut well, the following example will be worked out step by step:

Well Data

1. Physical Description

   a. Casing Size — 7" OD, 29#
   b. Tubing — 6300 ft of 3-1/2"OD UE 8rd

c.	Total Depth	7500 ft
d.	Perforations	6400 ft - 6900 ft
e.	Pump Setting Depth	6300 ft (100 ft above top perforations)
f.	Other	Corrosive Environment

2. Production Data

a.	Static Fluid Level (P_i)	2000 ft from surface
b.	Present Pumping Fluid Level (P_wf)	4000 ft from surface
c.	Present Producing Rate	3000 BFPD (1800 BOPD)
d.	BH Temperature	180°F
e.	Gas-Fluid Ratio	50
f.	Water-Oil Ratio	60
g.	Surface Discharge Pressure	100 psi
h.	Casing Pressure	0 psi (vented)
i.	Static Bottom Hole Pressure	1791 psi

3. Well Fluid Data

a.	A.P.I. gravity of oil	30°
b.	Specific gravity of oil	0.876
c.	Specific gravity of water	1.02

4. Power Supply

a.	Service Voltage	12,470 v.3Ø, $6H_z$
b.	Line capacity	Adequate

Assuming a consistent fluid gradient in the annulus, the producing and static liquid levels are proportional to the pumping fluid level (P_{wf}) and the static fluid level (P_i). Using the given production and fluid level data, an "Inflow Performance Relationship (IPR)" curve following the Vogel projection can be drawn as shown in Figure 2.

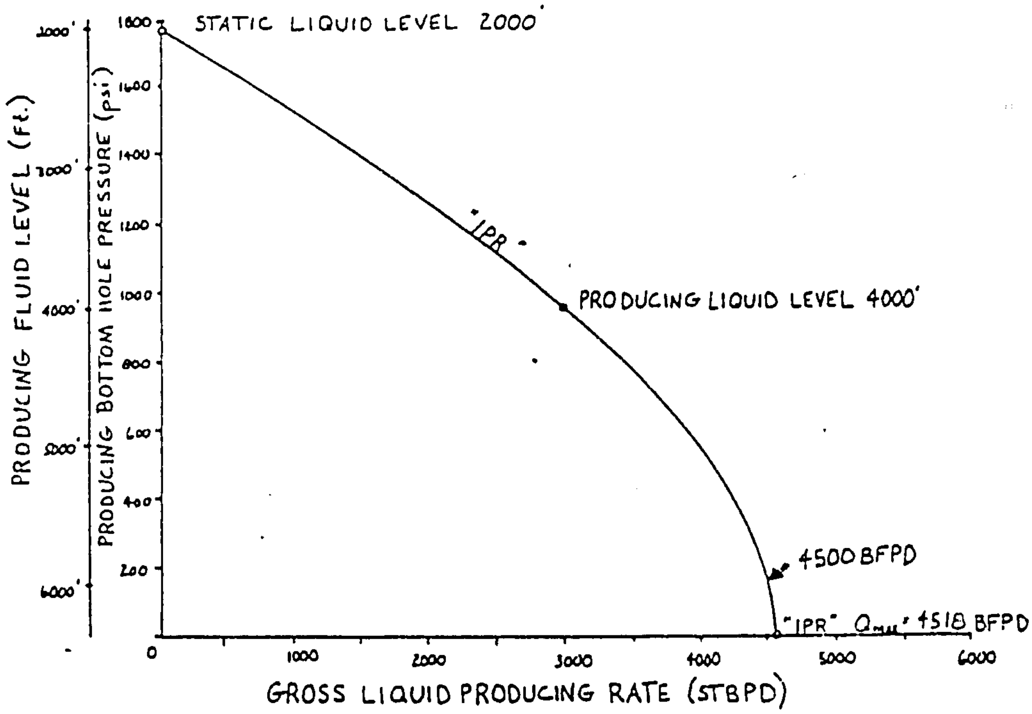
The q_{max} for this curve can be determined by solving the equation:

$$q/q_{max} = 1-0.2(P/P_s) - 0.8(P/P_s)^2$$

142

FIGURE 2

INFLOW PERFORMANCE RELATIONSHIP (IPR)



After the well data has been collected and plotted, the next step is to determine the well's production rate at the desired pump setting depth. This rate should be chosen from the IPR curve at a producing fluid level where there is some, but not an excessive amount of fluid over the pump.

From the IPR data, we would design for a production rate of 4,500 BFPD with a producing liquid level of 6,000 ft.

Total Dynamic Head (TDH)

$$TDH = h_d + f_t + P_d$$

h_d = 6,000 ft

f_t = calculated using standard correlation charts (from Hagen and Williams, C = 120) included in submersible pump catalogs from either REDA or CENTRILIFT.

$$f_t = \frac{6300\ ft}{1000\ ft} \times 53 \ \frac{\text{feet of loss}}{1000\ \text{ft of tubing}} = 334 \text{ feet}$$

P_d = discharge pressure head. Assume specific gravity of heaviest fluid to be pumped or 1.02

$$P_d = \frac{100\ psi \times 2.31\ psi/ft}{1.02} = 226\ ft$$

$$TDH = 6000\ ft + 334\ ft + 226\ ft = 6560\ ft$$

Pump Selection

After calculating the TDH and knowing the casing size, the pump type can be selected and number of stages determined. The casing size determines the series, or largest outside diameter of the pump and motor that can be used.

Assuming 7", 29# casing for this example and using a REDA Pump for illustration, (see Table 1 and Figure 3), the correct pump would be "G" type, 540 series (5.13 in. OD) pump.

1. For a desired rate of 4500 BFPD:

   Select G-150, Recommended Range 4400-6400 BFPD . Pump stages are designed to be in hydraulic balance near the peak efficiency. The recommended range on the pump curve defines these limits. Pump will operate in downthrust to the left of the recommended range or operate in upthrust to the right of the recommended range. Mechanical wear due to unbalance can shorten the pump life and the protector thrust bearing load will be significantly increased.

TABLE 1

TRW REDA ENGINEERING TABLES

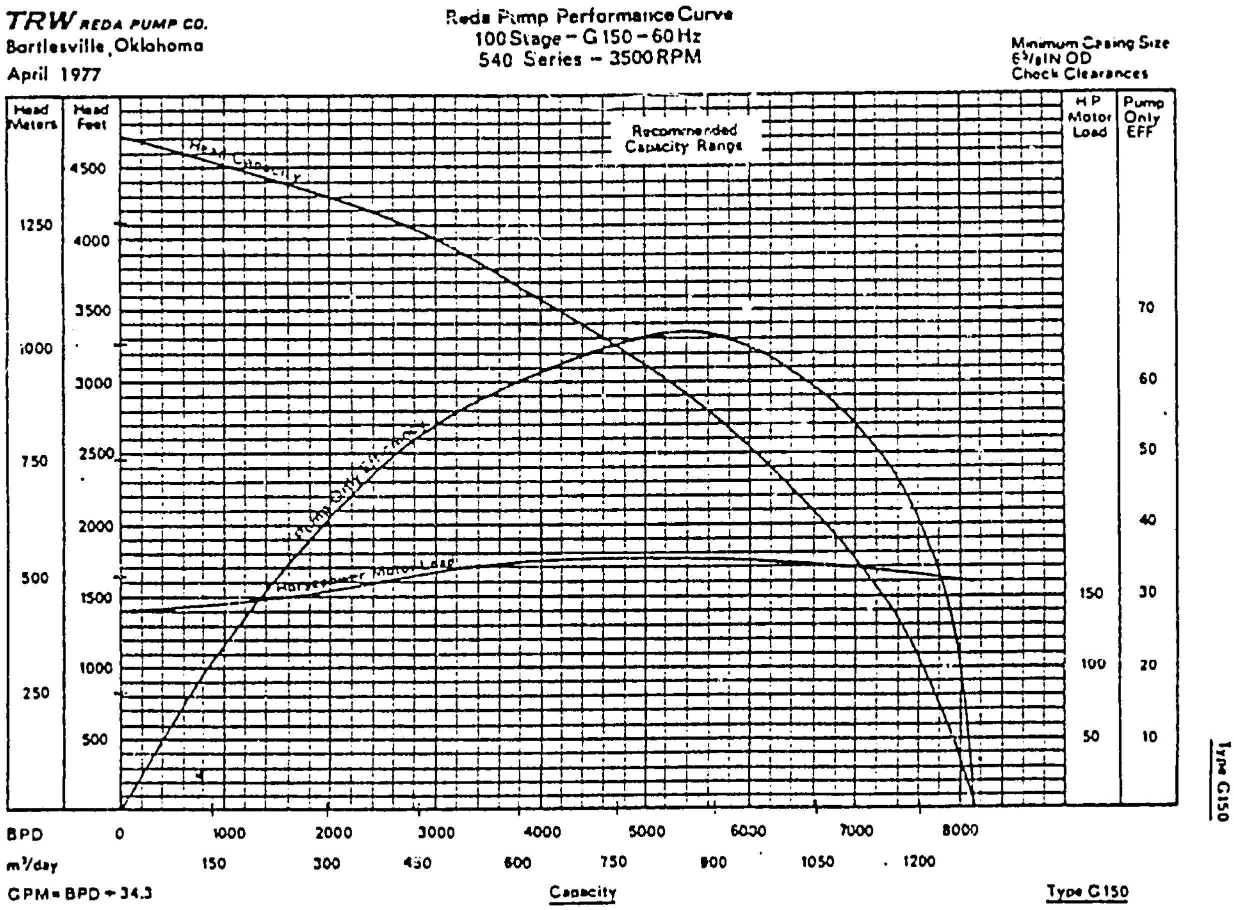
PUMPS 60 Hz 3500 RPM

SERIES	OUTSIDE DIAMETER (INCHES)	PUMP TYPE	*MAX BHP RATING FOR PUMP SHAFT	CAPACITY RANGE RECOMMENDED LIMITS (BPD)	(m³/D)
338	3.38	A-10	75	280-450	45-72
		A-14E	75	480-640	76-102
		A-25E	75	700-1060	112-168
		A-30E	75	900-1500	143-239
		A-45E	100	1200-1800	191-286
400	4.00	D-12	75	280-460	43-72
		D-13E	75	320-500	51-80
		D-15	75	400-600	64-95
		D-20	75	550-875	87-139
		D-26	100	780-1060	124-168
		D-40	100	950-1800	151-286
		D-51	100	1400-2000	223-318
		D-55E	100	1400-2400	223-382
		D-82	205	2100-3500	334-556
450	4.62	E-35E	130	1000-1500	159-238
		E-41E	130	1100-1750	175-278
		E-100	205	2900-4300	461-684
540	5.13	G-52E	205	1650-2350	262-374
		G-62E	205	1800-2700	286-429
		G-75P	205	2200-3200	350-509
		G-59E	205	2500-3600	397-572
		G-110	300	3200-4500	509-715
		G-150	300	4400-6400	700-1017
		G-180	300	5000-7250	795-1153
		G-220	300	6000-8000	954-1272
562	5.62	H-350	300	9200-15200	1462-2416
650	6.62	I-250	510	6500-9000	1033-1430
		I-300	510	8000-11500	1272-1828
675	6.75	J-400	510	13600-18000	2162-2862
		J-600	510	17000-24000	2702-3816
825	8.25	L-1050	1000	32000-42000	5087-6677
862	8.62	M-520	510	12000-24000	1908-3816
		M-675	510	19000-30000	3020-4770
950	9.50	N-1050	1000	24000-45000	3816-7154
1000	10.00	N-1500	1000	35000-55000	5564-8744

* 25% overload can be permitted where pump length is 20 feet or greater.

Note: Larger capacity pumps of larger horsepower available upon request.

FIGURE 3



TRW REDA PUMP CO.
Bartlesville, Oklahoma
April 1977
Reda Pump Performance Curve
100 Stage – G 150 – 60 Hz
540 Series – 3500 RPM
Minimum Casing Size
6⁵/₈IN OD
Check Clearances
Head Meters
Head Feet
Recommended Capacity Range
Head Capacity
Pump Only Efficiency
Horsepower Motor Load
HP Motor Load
Pump Only EFF
1250
1000
750
500
250
4500
4000
3500
3000
2500
2000
1500
1000
500
150
100
50
70
60
50
40
30
20
10
BPD 0 1000 2000 3000 4000 5000 6000 7000 8000
m³/day 150 300 450 600 750 900 1050 1200
GPM = BPD ÷ 34.3
Capacity
Type G150

2. From Table 2, read the desired pump output at 4500 BFPD and find:

Head per 100 stages = 3200 feet or 32.0 feet/stage
Horsepower per 100 stages = 178 HP or 1.78 HP/stage

3. Calculate Pump Stages and Horsepower Required:

$$\text{No. of Stages} = \frac{\text{TDH}}{\text{Head/Stage}} = \frac{6560 \text{ feet}}{32.0} = 205 \text{ Stages}$$

HP Required = No. of Stages x HP/Stage x Specific Gravity

HP = 205 Stages x 1.78 HP/Stage x 0.962 = 351 HP

4. Select Full-housing Pump and Motor

Pump = 205 Stage G-150

Motor = 360 HP 540 Series Tandem
1890 Volts, 120 amps

5. Select the motor voltage, cable type and switchboard that will minimize investment. Referring to the amp carrying capacity of cables, select size No. 1 cable. Referring to Figure 4 (Voltage Drop Curve), Cable No. 1 has a voltage drop of approximately 33 volts per 100 ft at 180°F at 120 amps. Therefore, the voltage drops for 6,400 ft of No. 1 cable is:

$$6{,}400 \text{ ft (Cable)} \times \frac{33 \text{ volts}}{1000 \text{ ft}} = 211 \text{ volt loss}$$

The required surface voltage will be:

$$V_s = \text{Surface Voltage} = 1890 \text{ volts} + \frac{6{,}400 \text{ ft} \times 33 \text{ volts}}{1000 \text{ ft}}$$

$$= 2101 \text{ Volts}$$

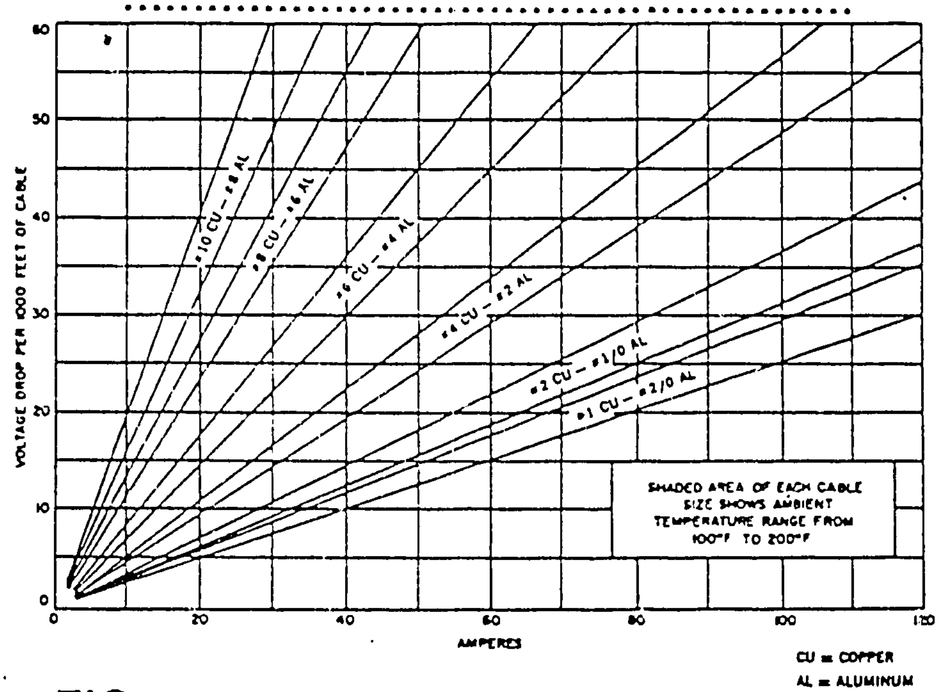


FIG. 4 - VOLTAGE DROP PER 1,000 FEET OF CABLE — CABLE SIZES FROM #1 TO #10

Check manufacturer tables for round cable clearance between tubing collars and well casing and select the type of cable which is appropriate. If the selected cable size cannot be used, it may be necessary to select a higher motor voltage which will reduce the amperage and permit use of a smaller cable size, or us flat cable.

Also, it may be advisable to change tubing sizes or to turn down the tubing collars. In determining the optimum cable size, consider the future equipment possibilities that may require larger cable size.

Switchboard Selection

Choose the switchboard based on maximum surface voltage, current, and horsepower rating with consideration given to future use.

Transformer Selection

First, calculate to total KVA required as follows:

$$KVA = \frac{2100 \text{ volts} \times 120 \text{ amps} \times 1.73}{1000} = 436 \text{ KVA}$$

(Several useful electrical conversion formulas are given in Table 2 for further reference.)

CONVERSION FORMULAS

TO FIND	DIRECT CURRENT	ALTERNATING CURRENT Single Phase	ALTERNATING CURRENT † Three Phase
Amperes When Horse Power (Input) is Known	$\frac{\text{H.P.} \times 746}{\text{Volts} \times \text{Efficiency}}$	$\frac{\text{H.P.} \times 746}{\text{Volts} \times \text{Efficiency} \times \text{P.F.}}$	$\frac{\text{H.P.} \times 746}{\text{Volts} \times 1.73 \times \text{Efficiency} \times \text{P.F.}}$
Amperes When Kilowatts is Known	$\frac{\text{KW} \times 1000}{\text{Volts}}$	$\frac{\text{KW} \times 1000}{\text{Volts} \times \text{P.F.}}$	$\frac{\text{KW} \times 1000}{\text{Volts} \times 1.73 \times \text{P.F.}}$
Amperes When kva is Known		$\frac{\text{kva} \times 1000}{\text{Volts}}$	$\frac{\text{kva} \times 1000}{\text{Volts} \times 1.73}$
Kilowatts	$\frac{\text{Amperes} \times \text{Volts}}{1000}$	$\frac{\text{Amps.} \times \text{Volts} \times \text{P.F.}}{1000}$	$\frac{\text{Amps.} \times \text{Volts} \times 1.73 \times \text{P.F.}}{1000}$
kva		$\frac{\text{Amps.} \times \text{Volts}}{1000}$	$\frac{\text{Amps.} \times \text{Volts} \times 1.73}{1000}$
Power Factor	(P.F.)	$\frac{\text{Kilowatts} \times 1000}{\text{Amps.} \times \text{Volts}}$ or $\frac{\text{KW}}{\text{kva}}$	$\frac{\text{KW} \times 1000}{\text{Amps.} \times \text{Volts} \times 1.73}$ or $\frac{\text{KW}}{\text{kva}}$
Horse Power (Output)	$\frac{\text{Amps.} \times \text{Volts} \times \text{Efficiency}}{746}$	$\frac{\text{Amps.} \times \text{Volts} \times \text{Efficiency} \times \text{P.F.}}{746}$	$\frac{\text{Amps.} \times \text{Volts} \times 1.73 \times \text{Efficiency} \times \text{P.F.}}{746}$

Power Factor and Efficiency when used in above formulas should be expressed as decimals.
† For 2-phase, 4-wire substitute 2 instead of 1.73.
† For 2-phase, 3-wire substitute 1.41 instead of 1.73.

TABLE 2

Three (3) 150 KVA single-phase transformers would provide the necessary KVA, but should the unit require resizing to a larger unit, the transformers would most likely be overloaded and require change out. Therefore, the recommended transformer selection is three 160 KVA single-phase transformers with 7,200/12,470 volts primary taps and 1900-2500 volts secondary taps.

This particular example illustrates the procedure for sizing a submersible pump for a high water-cut well. Other applications such as:

- Viscous production
- High GLR wells
- Surface injection booster

require different techniques for pump sizing.

Computer Design Approach

To optimize the selection of pumps, both REDA and CENTRILIFT have developed computer design programs. However, as with the hand calculation procedures, it is vital to provide pertinent well production details.

The software program will then select a pump configuration based on this production data. The program has three main design phases:

1. Input capacity Calculations

2. Tubing Pressure Drop

3. Pump Selection

FAILURE ANALYSIS

A knowledge of the operational success of similar type installations in an area can be very useful in helping to select equipment accessories and good operating procedures. Operating personnel can also contribute a great deal to the overall success in the performance of submersible pumping equipment. The main causes of failures include.

1. Excessive overload over an extended time period
2. Leaks in the seal section
3. Well Conditions - insufficient fluid movement, high temperature, corrosion, and abrasives in the fluid stream.
4. Bad or faulty installation
5. Switchboard troubles
6. Faulty Equipment
7. Worn out pump
8. Lightning
9. Bad electrical system

Because a submersible pumping system is quite sophisticated compared to other types of artificial lift equipment, it is especially important that every precaution be taken during the installation and operation to minimize the risk of failures.

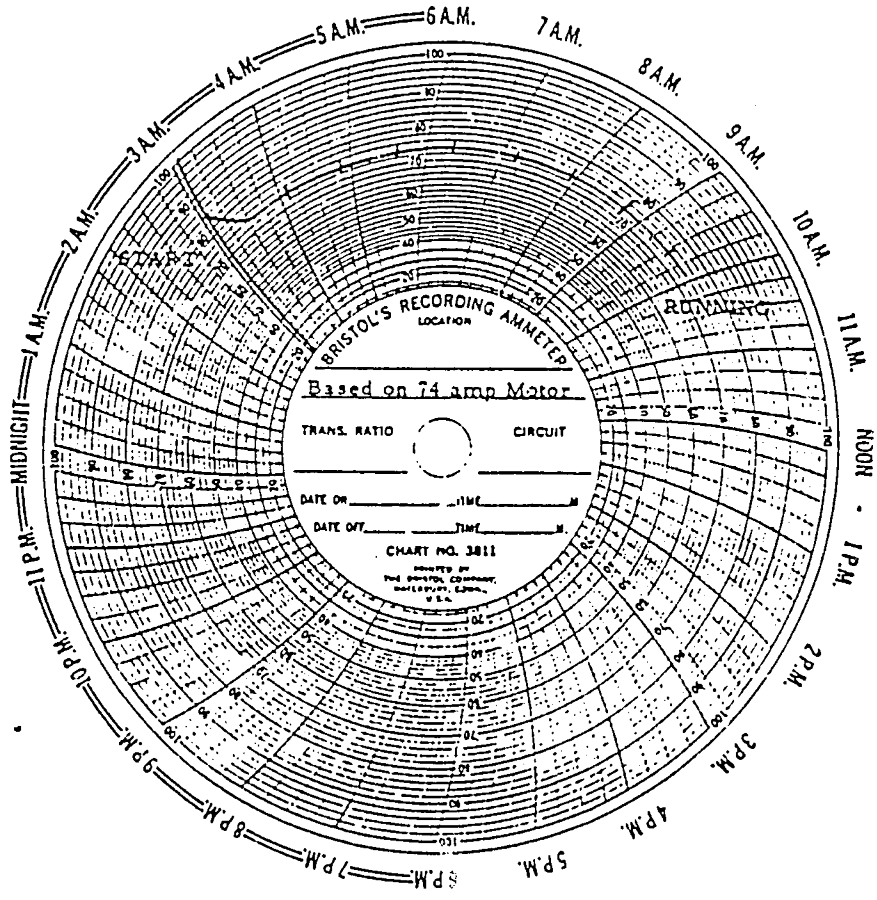
The usual cause of a premature failure for a properly designed unit is an unattended, correctable mechanical malfunction which leads to a downhole failure. It is, therefore, mandatory that each unit be properly and rigorously monitored to allow these malfunctions to be corrected before premature failure occurs.

One of the most valuable and least understood tools available is the recording ammeter. The ammeter chart, like a physician's electrocardiogram, is a recording of the heartbeat of the submersible electrical motor. Proper, timely and rigorous analysis of amp charts can provide valuable information for the detection and correction of minor operational problems before they become costly major ones.

Three examples of these charts are shown in Figures 5, 6 and 7. Additional suggestions for "Troubleshooting" are available from most manufacturers.

FIGURE 5

TYPICAL AMMETER CHARTS



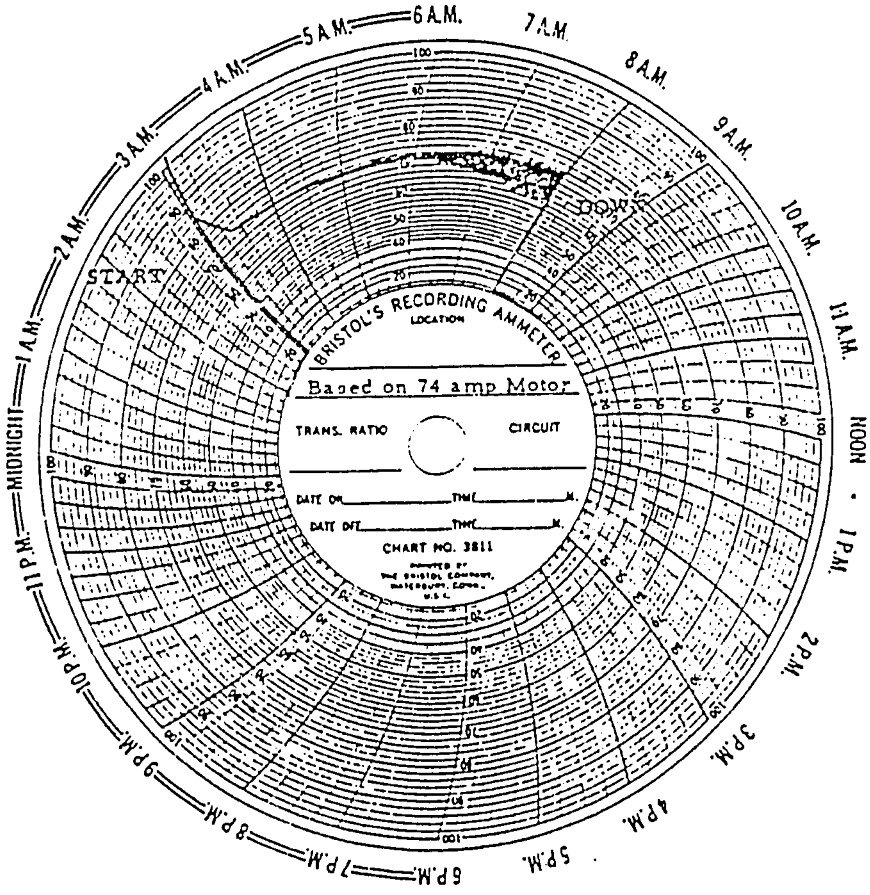
The start up on this chart indicates a normal draw down and continuing good production rate. Very little, or no gas or good gas separation, is indicated by the smooth steady current holding at 74 amperes.

The "blips" shown at regular intervals are caused by power fluctuations. They are probably due to periodically starting another heavy electric load on the power system.

REDA PUMP COMPANY — BARTLESVILLE, OKLAHOMA
A DIVISION OF TRW INC.

FIGURE 6

D. TYPICAL AMMETER CHARTS



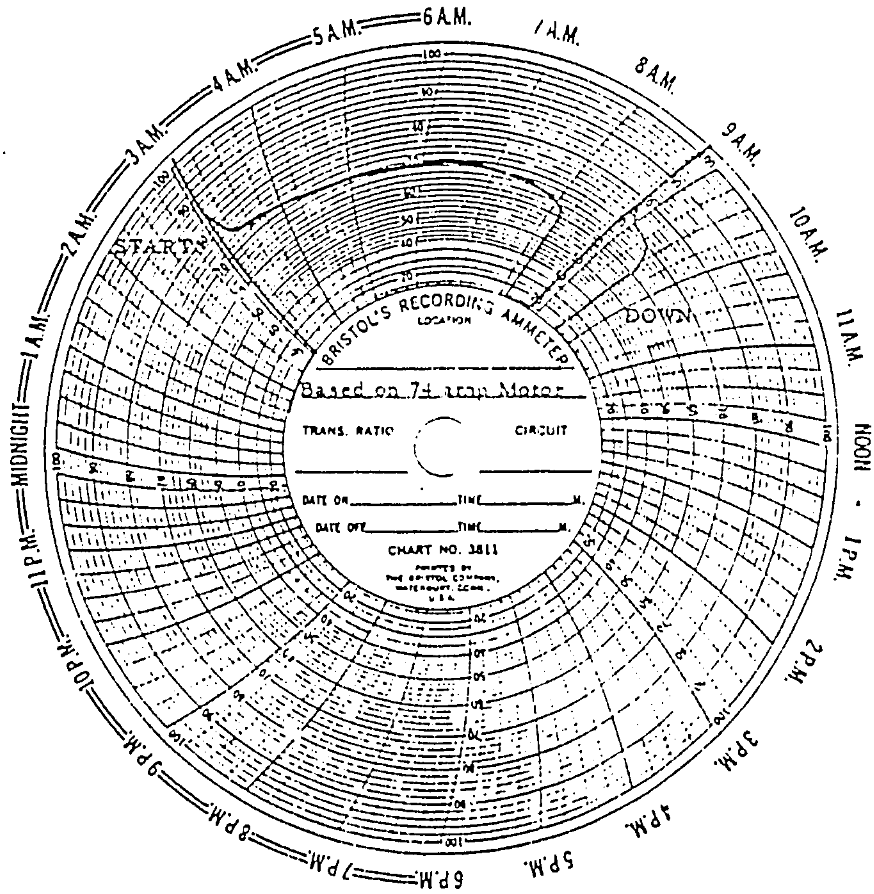
The amperage on this chart indicates that after a normal start, production rate goes into a slow decline, and gradually becomes very gassy; the unit pumped off or gas locked at 8 a.m. and went off on the undercurrent relay. The gas bubble point is shown to be very near the pump.

This condition can often be remedied by lowering the pump in the well a few more joints. Ineffective gas separation is indicated and/or pumped-off condition.

REDA PUMP COMPANY — BARTLESVILLE, OKLAHOMA
A DIVISION OF TRW INC.

FIGURE 7

D. TYPICAL AMMETER CHARTS



From a static level this well pumped off in 5 hours 10 minutes. After allowing 35 minutes build up time, the pump started again and ran about 45 minutes, then pumped off again. Indication: The well is not prolific enough for this large a pump. Possible remedies: Well work-over (frac, acid, re-perforating, clean out, etc.) in attempt to bring more fluid into well. If work over fails to produce more fluid, the need for a smaller pump is clearly indicated. This could result in continual running time, higher efficiency, lower pumping unit cost, better overall economy.

REDA PUMP COMPANY — BARTLESVILLE, OKLAHOMA
A DIVISION OF TRW INC.

RECENT INNOVATIONS

Several unique submersible installations which are gaining in use are:

- Cable Suspended Systems
- Variable Speed Systems
- Tapered Submersible Pumps
- Rotary Gas Separator

Cable Suspended Systems - offer attractive economics when considering crude oil production either for offshore platforms, or in field locations remote from normal sources of supply and service.

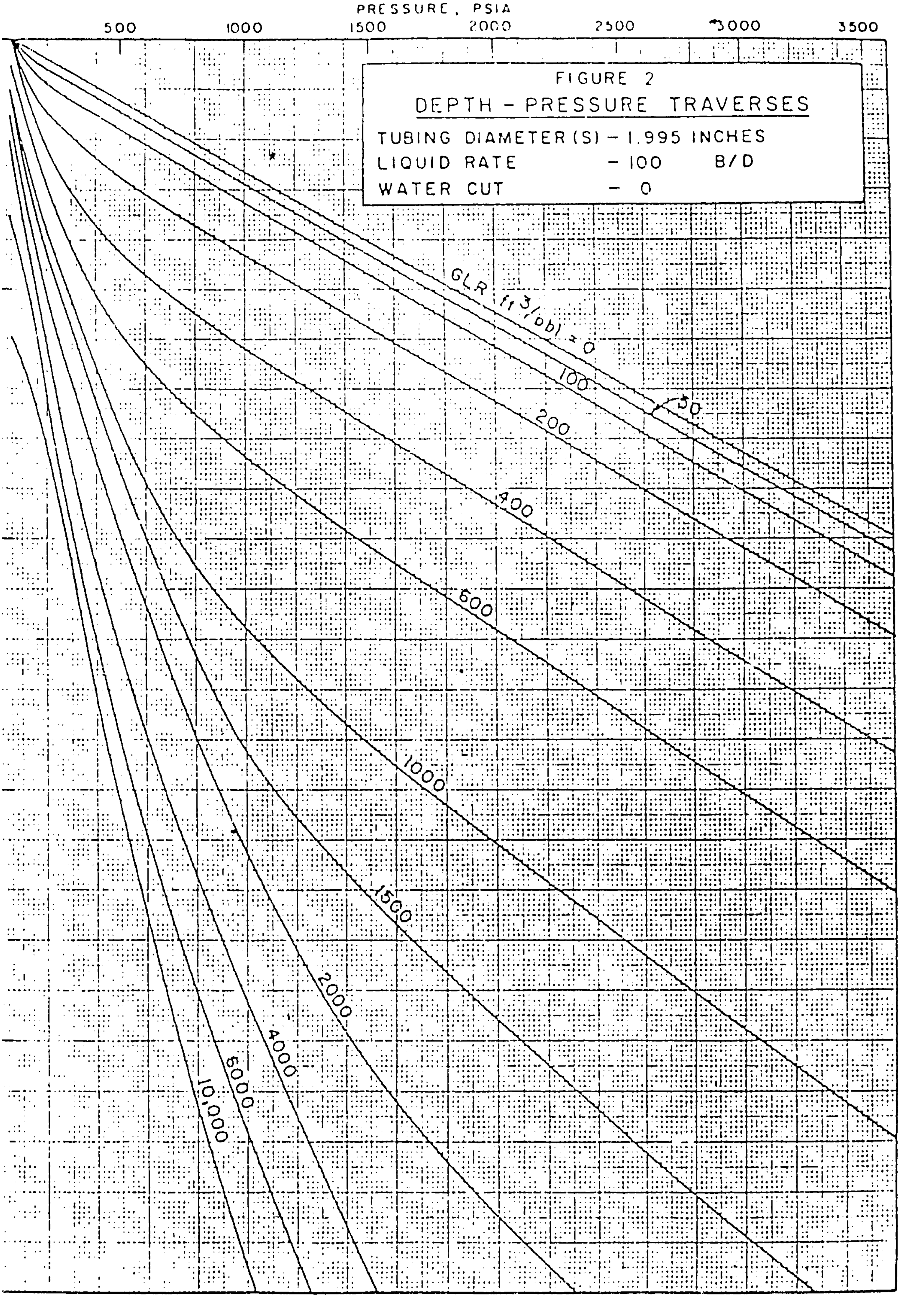
Variable Speed Systems - provides a method using existing downhole equipment to obtain accurate productivity data on wells requiring larger electric submersible pumps.

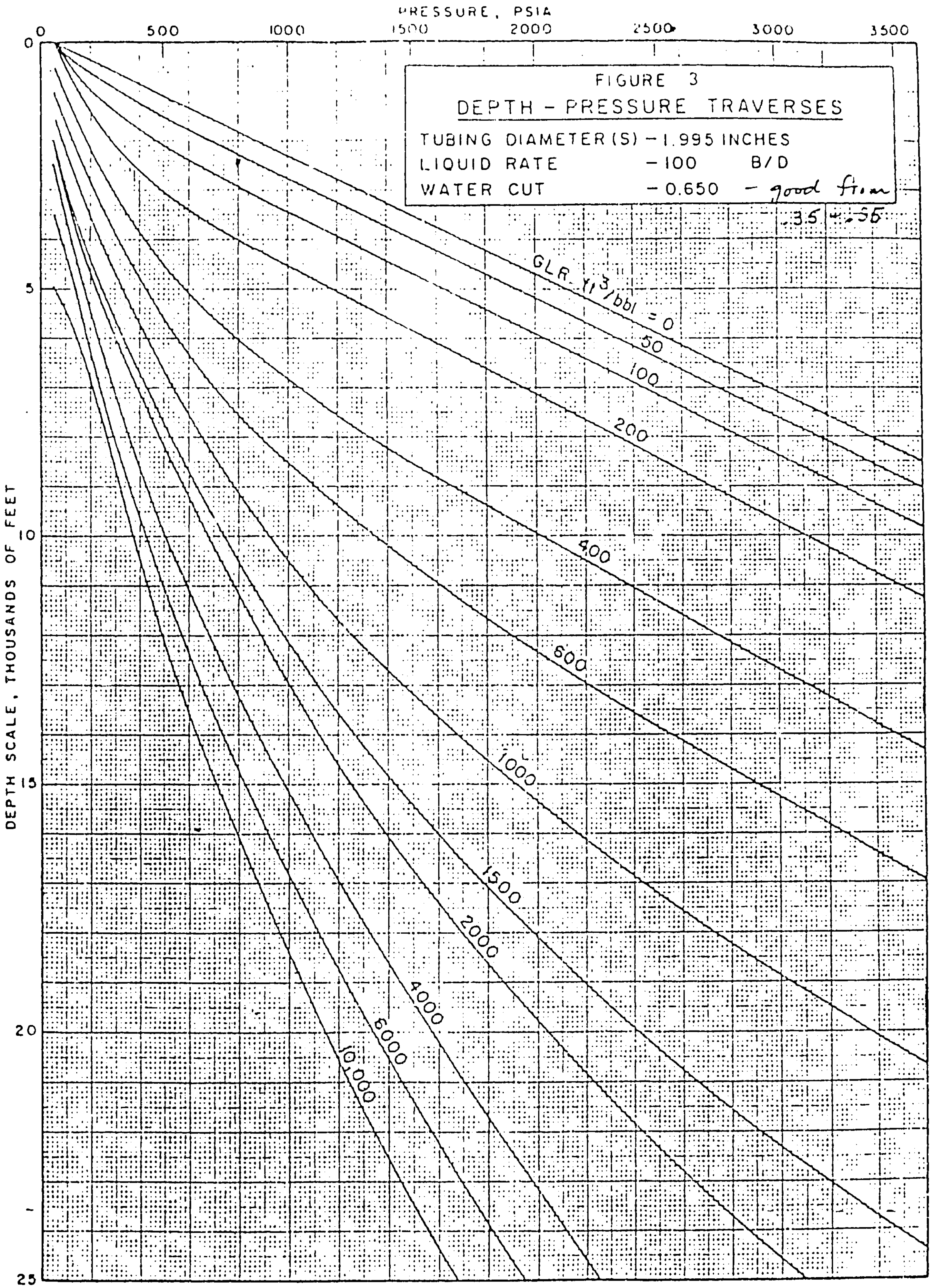
Tapered Submersible Pumps - will allow production of high gas-liquid ratio reservoirs by putting larger capacity pump stages on bottom to compress the fluid and smaller stages above to boost the fluid to surface within the optimum capacity range.

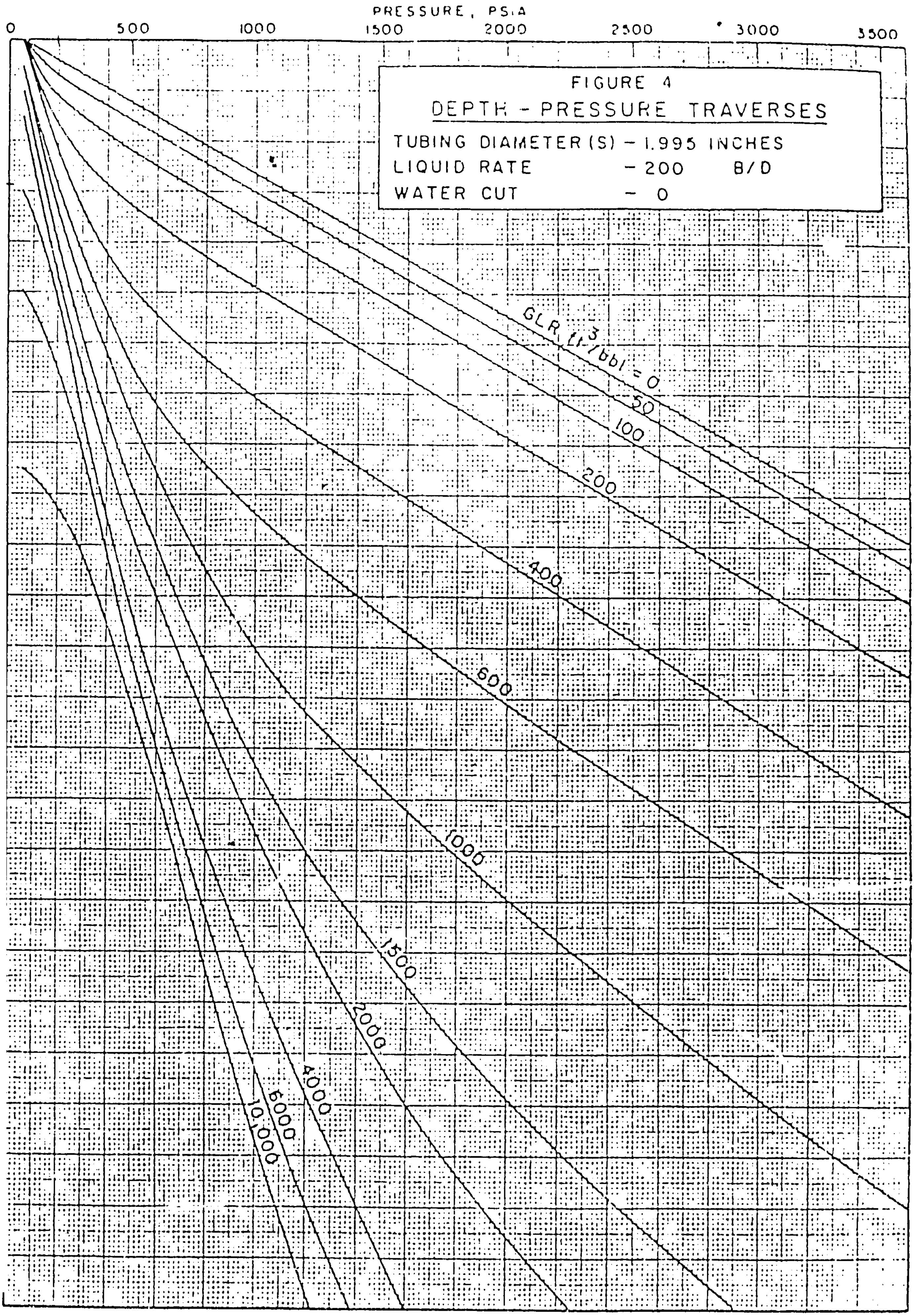
Rotary Gas Separator - recently offered by CENTRILIFT, is designed to remove most of the free gas before the production reaches the pump intake, eliminating the adverse effects on pump efficiencies, cavitation, and unstable motor loads.

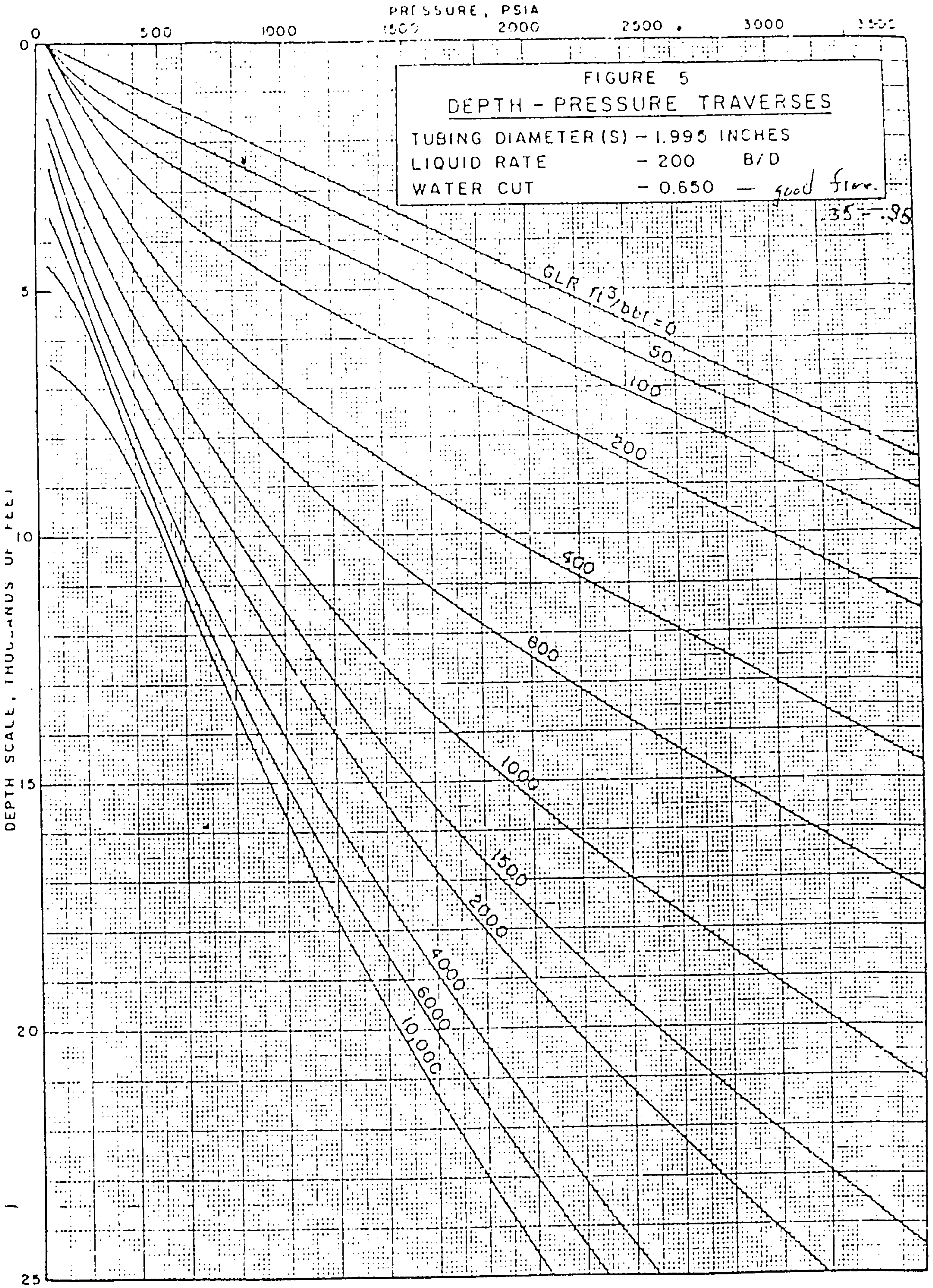
Several SPE papers and trade journal articles on these and other submersible pumping related subjects are included for further background and reference. Also refer to the Submersible Pump Handbook provided by CENTRILIFT and other manufacturer references for additional details.

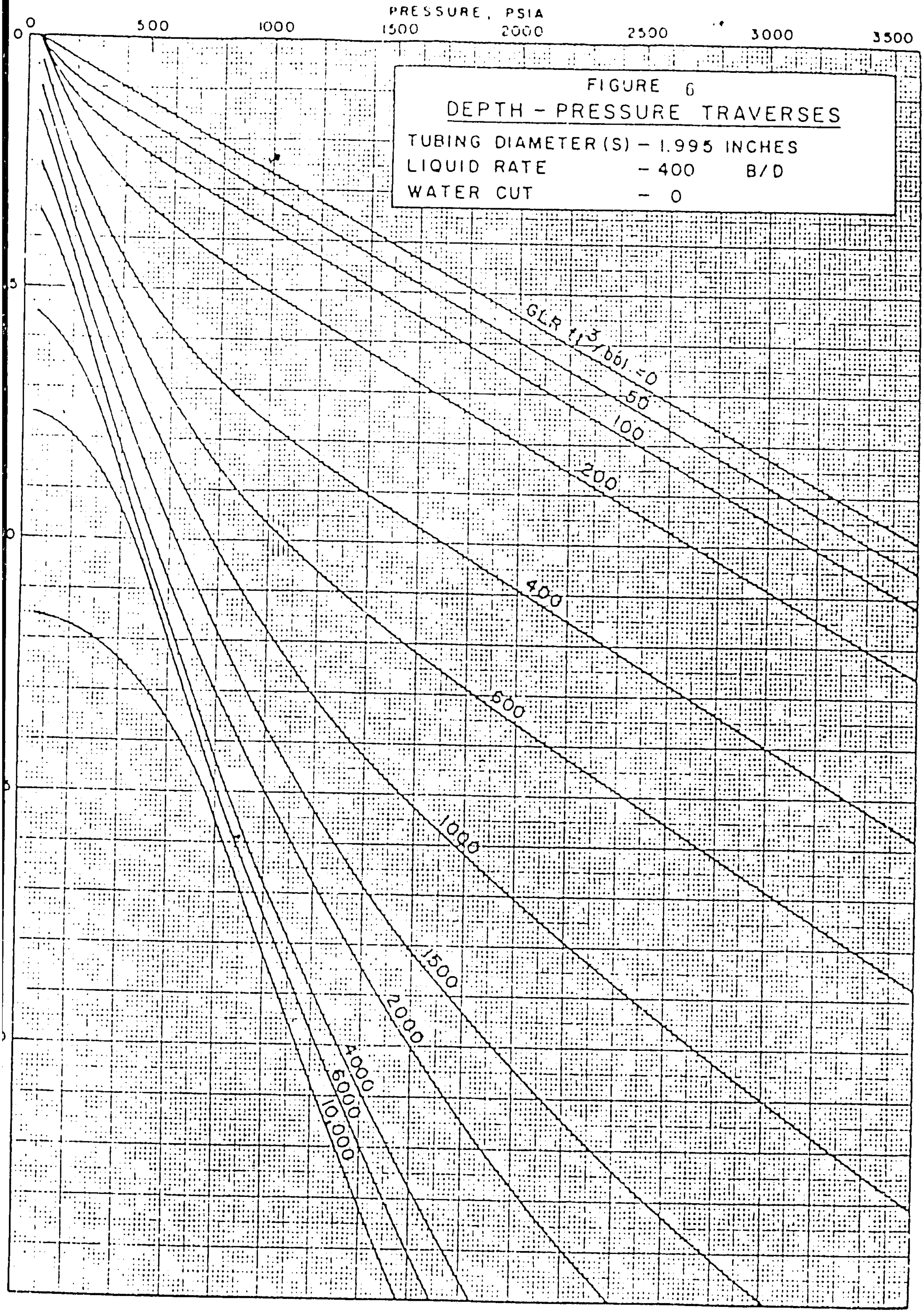
Pressure Traverse Charts


PRESSURE, PSIA
500
1000
1500
2000
2500
3000
3500
FIGURE 2
DEPTH - PRESSURE TRAVERSES
TUBING DIAMETER (S) - 1.995 INCHES
LIQUID RATE - 100 B/D
WATER CUT - 0
GLR ft3/bbl = 0
50
100
200
400
600
1000
1500
2000
4000
6000
10,000

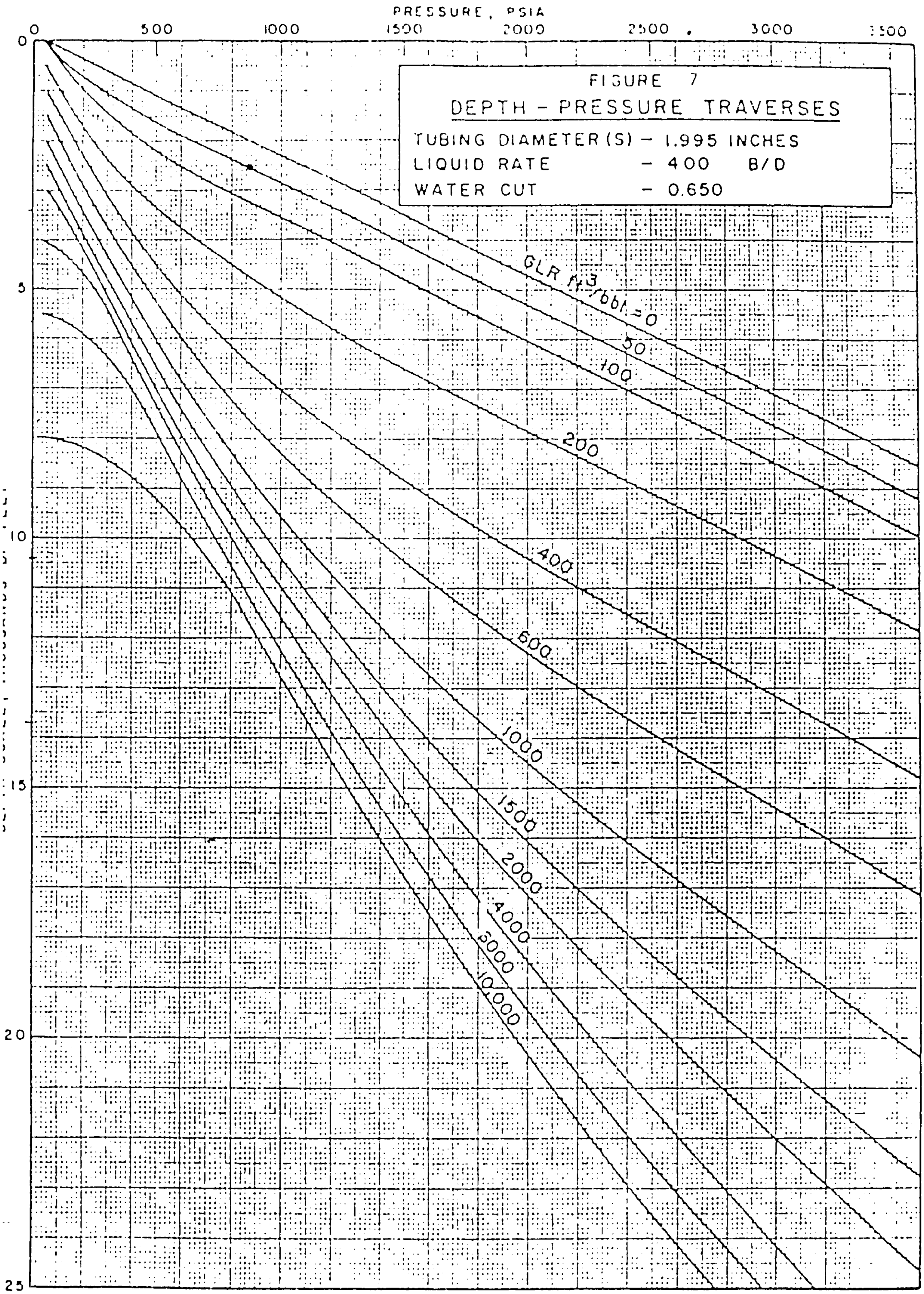

PRESSURE, PSIA
0
500
1000
1500
2000
2500
3000
3500
FIGURE 3
DEPTH - PRESSURE TRAVERSES
TUBING DIAMETER(S) - 1.995 INCHES
LIQUID RATE - 100 B/D
WATER CUT - 0.650 - good from
.35 - .56
GLR, ft³/bbl = 0
50
100
200
400
600
1000
1500
2000
4000
6000
10,000
DEPTH SCALE, THOUSANDS OF FEET
5
10
15
20
25

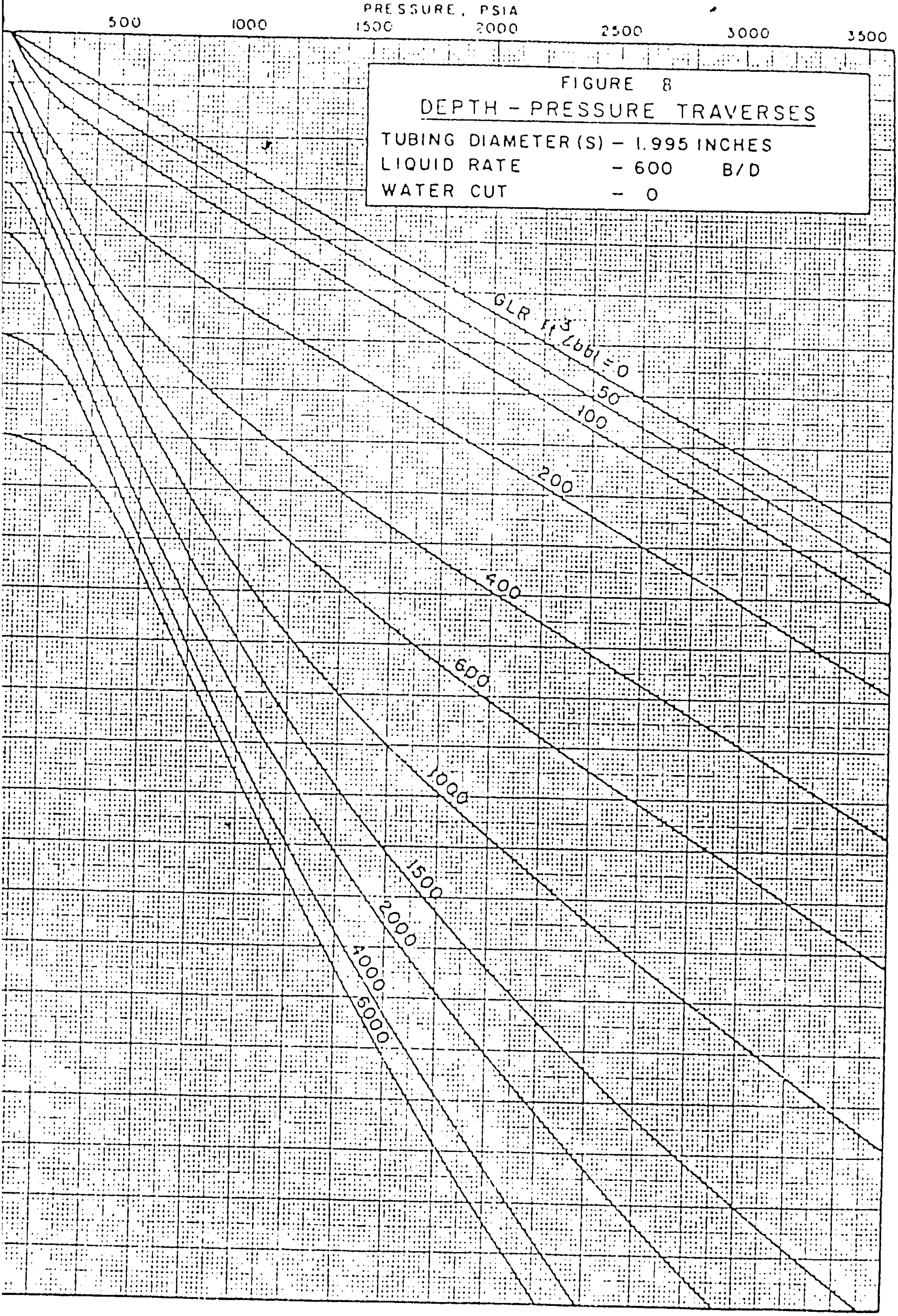

PRESSURE, PSIA
0
500
1000
1500
2000
2500
3000
3500
FIGURE 4
DEPTH - PRESSURE TRAVERSES
TUBING DIAMETER (S) - 1.995 INCHES
LIQUID RATE - 200 B/D
WATER CUT - 0
GLR ft³/bbl = 0
50
100
200
400
600
1000
1500
2000
4000
6000
10,000


PRESSURE, PSIA
0
500
1000
1500
2000
2500
3000
3500
FIGURE 5
DEPTH - PRESSURE TRAVERSES
TUBING DIAMETER (S) - 1.995 INCHES
LIQUID RATE - 200 B/D
WATER CUT - 0.650 — good flow.
.35 - .95
GLR ft³/bbl = 0
50
100
200
400
600
1000
1500
2000
4000
6000
10,000
DEPTH SCALE, THOUSANDS OF FEET
0
5
10
15
20
25



PRESSURE, PSIA
0
500
1000
1500
2000
2500
3000
3500
FIGURE 6
DEPTH - PRESSURE TRAVERSES
TUBING DIAMETER (S) - 1.995 INCHES
LIQUID RATE - 400 B/D
WATER CUT - 0
GLR ft3/bbl =0
50
100
200
400
600
1000
1500
2000
4000
6000
10,000


PRESSURE, PSIA
0
500
1000
1500
2000
2500
3000
3500
0
5
10
15
20
25
FIGURE 7
DEPTH - PRESSURE TRAVERSES
TUBING DIAMETER (S) - 1.995 INCHES
LIQUID RATE - 400 B/D
WATER CUT - 0.650
GLR ft3/bbl=0
50
100
200
400
600
1000
1500
2000
4000
5000
10,000


PRESSURE, PSIA
500
1000
1500
2000
2500
3000
3500
FIGURE 8
DEPTH - PRESSURE TRAVERSES
TUBING DIAMETER (S) - 1.995 INCHES
LIQUID RATE - 600 B/D
WATER CUT - 0
GLR ft3/bbl = 0
50
100
200
400
600
1000
1500
2000
4000
6000

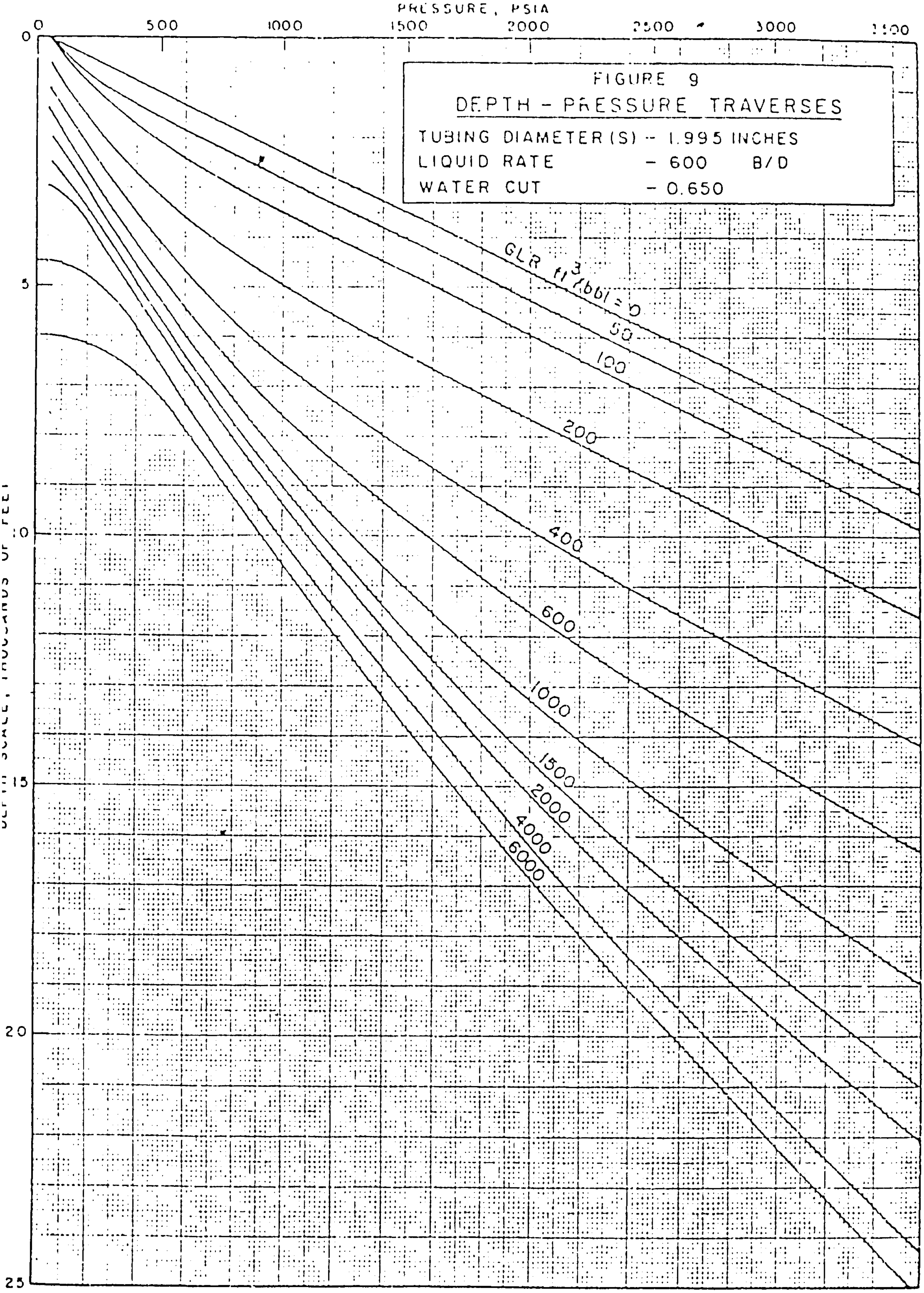


FIGURE 9
DEPTH - PRESSURE TRAVERSES
TUBING DIAMETER (S) - 1.995 INCHES
LIQUID RATE - 600 B/D
WATER CUT - 0.650
PRESSURE, PSIA
0
500
1000
1500
2000
2500
3000
3500
DEPTH SCALE, THOUSANDS OF FEET
5
10
15
20
25
GLR ft³/bbl = 0
50
100
200
400
600
1000
1500
2000
4000
6000

Resumé

DR. R. EUGENE COLLINS

For more than thirty years, Dr. R. Eugene Collins has been involved with research, teaching, and consulting in petroleum engineering, reservoir engineering, and well completion research. He has worked in the research laboratories of three major oil companies and is currently a consultant to Amoco Production Company, Schlumberger-Doll Reserach Laboratory, and the U.S. National Bureau of Standards. His numerous technical publications include a widely known textbook, Flow of Fluids through Porous Materials, now in its second printing. His book has been translated into both Russian and Japanese.

Dr. Collins received his B.A. degree in physics from the University of Houston and both his M.S. and Ph.D. in physics from Texas A & M. For twenty years he was a Professor of Physics at the University of Houston. Now he is the Frank W. Jessen Professor of Petroleum Engineering at the University of Texas at Austin. For two years he also has been president of his own consulting firm, Research and Engineering Consultants.

Among Dr. Collins numerous memberships in professional and honary societies are memberships in Sigma Pi Honor Society, the Society of Petroleum Engineers, and the honary society Pi Epsilon Tau. Dr. Collins is a Registered Professional Engineer in Rexas.

164